A STUDY OF THERMAL COMFORT

AND COST EFFECTIVENESS

OF STRATUM VENTILATION

by

Alan Ming-Lun Fong

(M. L. Fong)

This thesis submitted

in partial fulfillment

of the award of

Doctor of Philosophy

of

De Montfort University

December 2014

(Amended version incorporated with minor corrections following by viva voce held on 1st June 2015)

Abstract

This study focuses on thermal comfort and cost effectiveness of stratum ventilation in subtropical HKSAR.

The need for studying thermal comfort with various air distribution strategies becomes a significant issue recently due to climate change, increasing energy prices and the governmental energy efficiency policy. Stratum ventilation, with air supplied at breathing level, can provide satisfactory thermal comfort at a relatively elevated indoor temperature in which less energy use is consumed. It seems that only limited studies on the evaluation of neutral temperature, which is a condition of neither slightly warm nor slightly cool, are supported by actual human comfort surveys. Moreover, study on the related thermal comfort and cost effectiveness as other paradigms in comparison with the mixing and displacement air distribution design is rare.

New environmental chamber of laboratory-based air-conditioning systems has been developed for investigating the actual benefit of cost effectiveness and balance of thermal comfort satisfaction with the stratum air distribution strategy under subtropical climates. ASHRAE 7-point questionnaires have been collected from human comfort tests so as to estimate the neutral temperature of stratum ventilation in comparison with mixing and displacement ventilation at pre-set conditions. The neutral temperatures of HKSAR people under the mode of mixing, displacement, stratum, modified-stratum-1, modified-stratum-2, and modified-stratum-3 are found to be 24.6℃, 25.1℃, 25.6℃, 26.0℃, 27.1℃ and 27.3℃ at 10 ACH respectively, which become 24.8℃, 25.3℃, 26.6

℃, 27.4℃, and 27.9℃ at 15 ACH respectively.

Life cycle assessment results in 10 service year indicate that 7.37% and 7.32% of cost reduction, and 14.52% and 11.91% of greenhouse gas emission reduction in stratum ventilation by comparing with mixing and displacement ventilation. As a result, stratum ventilation should be the best option on both of cost reduction, and less carbon emission in small-to-medium size air-conditioned space of new building and retrofitting existing works.

Keywords:
Stratum ventilation; Thermal comfort; Neutral temperature; Uniformity; Cost effectiveness; Life cycle assessment; Energy.

Acknowledgements

The author would like to give my hearty gratitude to my two first supervisors Professor Victor Ian Hanby (October 2010 to December 2014) and Dr. Rich Greenough (January 2015 to June 2015), not just for his continual inspiration and encouraging response to my research works all along these four and a half years of part-time study, but also for his life impact of the enthusiasm and commitment in research.

I would also thanks to my two second and local supervisors Dr. John Zhang Lin and Dr. Kwong-Fai Fong for his continual advice on the research progress and matters, so that I would not be deflected from the priority of study in the demanding daily working life in City University of Hong Kong (CityU), and keep on the momentum of my study.

Sincere thanks are expressed to all of my colleagues in Division of Building Science and Technology, and my members of the Building Energy and Environmental Technology Research Unit for their encouragement and sharing of valuable experience. It is with great indebtedness that I wish to thank all of technical staff, research assistant and students at the environmental chamber for their assistance in conducting the data of human tests and experimental measurements. I will never forget the friendly research atmosphere coming from many kind and interesting people for their technical support.

The acknowledgement also includes my employer, the CityU to provide the environmental chamber and all necessary setup during my research study. I express my gratitude also to Mr. Yong Cheng as final year PhD students at CityU and Dr. Ting Yao for consolidating their collected data with my first set of data, described in Chapter four.

I am forever grateful to my beautiful wife Helen Po-Lin Sze for her love, inspiration, and patience. I am indebted to my parents for their infinite love and support from beginning to end. To my Lord Jesus Christ, I give thanks for being with me from the absolute beginning and always answering my prayers.

The last but not the least, I thank once again for my first my first supervisors Professor Vic Hanby and Dr. Rick Greenough, as well as my two second and local supervisors Dr. Kwong-Fai Fong, Dr. John Zhang Lin provided me with selfless guidance, mentorship, inspiration, and encouragement throughout these four and a half years of part-time study.

Thank you all of your support!

Contents

List of Tables

List of Figures

Figure	Description	Page

Chapter 3

Nomenclature

Variables	Description	Unit
$A(0)$	initial area of jet	m^2
$Ar(0)$	Archimedes number at the inlet	-
$B(0)$	Specific buoyancy flux	m^4/s^3
C_v	Volumetric specific heat of air	$kJ/(m^3.K)$
E	Entrainment in the jet	m^2/s
$E_{Airside}$	Annual energy saving in air side equipment	MJ
$E_{Waterside}$	Annual energy saving in water side equipment	MJ
E_{OM}	Annual operation and maintenance energy difference	MJ
g	Acceleration of gravity	m/s^2
H	Distance between floor and the center of the supply	m
h	Height of supply diffuser	m
l_T	Thermal length	m
L	Characteristic length of supply air terminal	m
L_{Traj}	Length of trajectory	m
m	Specific momentum flux	m^3/s
P_f	Fan power	kW
P_c	Refrigeration power	kW
P_t	Total power	kW
PMV	Predicted mean vote	-
PPD	Predicted percentage dissatisfied	-
Q	Cooling power	kW
q	Volumetric flow rate	m^3/s
Re	Reynolds number	-
ΔR	Uncertainty	-
$S\Phi$	Source term of general scalar Φ governing equation	-
s	Distance along trajectory	m
t	Time	s
t_x	Local airstream dry-bulb temperature	^oC

Variable	Description	Unit
t_c	Average room dry-bulb temperature	^oC
T_{cl}	Cloth temperature	^oC
T_p	Temperature of the occupied zone	^oC
T_i	Indoor temperature	^oC
T_m	Mixed air temperature	^oC
T_0	Outdoor air temperature	^oC
$T_{s\ or}\ T_{in}$	Supply air temperature	^oC
T_r	Mean radiant temperature	^oC
T_{So}	Overall thermal sensation	-
T_R	Temperature of the exhaust air	^oC
T_u	Turbulent intensity	%
T	Temperature in room	K
T_{in}	Supply temperature	K
ΔT	Temperature difference between T and T_{in}	K
U	Velocity	m/s
U_{in}	Supply velocity	m/s
U_F	Supply velocity, face plate, convex hull of holes	m/s
U_H	Supply velocity, face plate hole	m/s
U_N	Supply velocity, nozzle	m/s
u	Air velocity	m/s
u_0	Initial jet velocity	m/s
u_m	Centerline velocity at distance x from supply	m/s
v	Air flow rate per kW cooling	m^3/kJ
V	Supply volume flow rate	m^3/s
V_x	Local airstream centerline speed	m/s
W	External work	W/m^2
w	Weight of subject	kg
X	Throw	m
x	Distance from supply	m

Variable	Description	Unit
y	Vertical distance normal to wall direction	m
$Z_{Airside}$	Annual reduction of GHG emission from the local utility service in airside system compared with MV	MJ
$Z_{Waterside}$	Annual reduction of GHG emission from the local utility service waterside system compared with MV	MJ
Ω_{AHU}	GHG emission during the complete manufacturing process of the air handling unit	MJ
$\Omega_{Installation}$	GHG emission for all necessary accessories of different ventilation installed on site	MJ
$\Omega_{Transporation}$	GHG emission due to the transportation of the system components from factory to installation site	MJ
Σ_{AHUPMV}	Embodied energy of the air handling unit from Cradle-to-gate	MJ
$\Sigma_{Installation}$	Embodied energy of all accessories of different ventilation system erected on site	MJ
$\Sigma_{Transportation}$	Embodied energy of transportation of the different ventilation system from the factory to installation site	MJ

Greek symbols

Φ	Corresponding quantity in general governing equation	-
$\Gamma\Phi$	Effective diffusion coefficient	-
μ	Dynamic viscosity of air	kg/(m·s)
μt	Turbulent viscosity of air	kg/(m·s)
ε	Dissipation rate of the kinetic energy of turbulence	m^2/s^3
εt	Efficiency of heat removal	-
θed	Effective draft temperature	K
θeds	Effective draft temperature for stratum ventilation	K
ρ	Air density	kg/m^3
$\Delta\rho$	Difference of supply and indoor air density	kg/m^3

Variable	Description	Unit
β	Fraction of Carnot COP of chiller	-
$\beta 0$	Intercept	-
$\beta 1$	Coefficient	-
υ	kinematic viscosity	m^2/s
u_τ	Shear stress velocity	m/s

List of Abbreviations

AFFL	Above finishing floor level
ASHRAE	American society of heating, refrigerating and air-conditioning engineers, inc.
AST	Actual mean thermal sensation votes
ACH	Air changes per hour
ADPI	Air diffusion performance index
AC	Alternative case
BC	Basic case
BEC	Code of Practices for building energy code
BMI	Body mass index
BMS	Building management system
BSA	Body surface area
CLCD	Chinese life cycle database
CFD	Computational fluid dynamics
CMLCA	Chain Management by Life Cycle Assessment
COP	Coefficient of performance
DDG	Double deflection grille
DNS	Direct Numerical Simulation
DO	Discrete ordinate
DV	Displacement ventilation
EAC	Code of Practices for Energy audit code
EDT	Effective draft temperature
EDTS	Effective draft temperature for stratum ventilation
EMSD	Electrical and mechanical services department
ELCD	European life cycle database
EPD	Environmental protection department
EPBP	Energy payback period
FVM	Finite volume method
GHG	Greenhouse gas

GPBP	Greenhouse gas payback period
HKSAR	HKSAR special administration region
IAQ	Indoor air quality
IC	Initial cost
ICE	Inventory of carbon and energy
LACI	Local air change index
LCA	Life cycle assessment
LCC	Life cycle costing
LCI	Life cycle inventory
LES	Large eddy simulation
LMAA	Local mean age of air
MC	Maintenance cost
MoE	Ministry of the environment
MV	Mixing ventilation
NPV	Net present value
PV	Present value
PDD	Predicted percentage of dissatisfied
PMV	Predicted mean vote
OC	Operation cost
RAL	Return air louvre
RH	Relative humidity
RNG	Re-Normalization group
SAG	Supply air grille
SV	Stratum ventilation
SV-1	Modified-stratum-1 ventilation
SV-2	Modified-stratum-2 ventilation
SV-3	Modified-stratum-3 ventilation
TRNSYS	TRaNsient System simulation program

CHAPTER 1: INTRODUCTION

1.1 Thermal comfort and energy conservation

The earth's climate is rapidly changing which may be caused by misuse of energy for human activities. The optimisation of the energy use becomes an urgent matter for each country in order to minimise the carbon emission and to satisfy thermal comfort perception. The purpose of maintaining a thermally comfortable environment is not only to satisfy people's desire to achieve thermal neutrality in which neither warmer nor cooler surroundings is preferred, but also to optimize our energy consumption. Reducing the energy consumed by central air-conditioning system can mitigate carbon emission and minimise the climate change effect. Several East Asia governments have made proactive moves by issuing new guidelines on temperatures in air-conditioned premises in summer, including recommended room condition of 25.5℃ by the Electrical and Mechanical Services Department (EMSD) of the HKSAR government, 26℃ by the National Development and Reform Commission of the Chinese State Council, 27℃ by the Office of President in Taipei, 26-28℃ by the Ministry of Knowledge and Economy of Korea, and more radically, 28℃ by the Ministry of the Environment (MoE) of Japanese cabinet. This kind of policy aims at maintaining sustainability of our planet, but what is the maximum temperature that people can sustain in an air-conditioned room if thermal comfort is not sacrificed? It is important to determine the thermal neutral temperature of different air distribution methods to screen out which system(s) can achieve thermal comfort in warm condition with good energy performance, as well as better cost effectiveness achievement.

1.2 Air distribution methods

Mixing ventilation is the commonly used conventional air distribution method in HKSAR nowadays. Air supply terminals are ceiling-mounted and significant fan energy should be consumed owing to the distance between the terminals and the occupants if a higher air movement in the occupied zone is required. The system is difficult to direct supply air horizontally in the occupied zone. For displacement ventilation, the air for breathing zone at around 1.1 m height is transported by the boundary layer around an occupant's body at seating position, and the air quality is a weighted average of the air quality in the room from the floor level transferred to breathing zone. Because the airflow is thermally driven, horizontal air movement will disrupt the thermal plume around an occupant and therefore defeat its working mechanism. Jackman (1990) recommended a mean air speed of <0.25 m/s in summer for the comfort of sedentary occupants under displacement ventilation to form a practical design criterion.

Lin et al. (2005, 2009) proposed an air distribution method – stratum ventilation - to cope for higher room temperature and air movement serving medium size air conditioned area in comparison with conventional air distribution method. Air is supplied horizontally by means of wall mounted air diffusers to the head–chest level/breathing zone. Personalized and task station ventilation also satisfies aforementioned two requirements under warm conditions. Personalized ventilation and stratum ventilation share similar principles. Those can fully make use of the cooling effect, low temperature, and air movement, of the supply air. Stratum ventilation, when compared with personalized ventilation, is less flexible. However the flexibility can be improved by adopting in-situ adjustable air terminals developed specifically for stratum ventilation, whereas the advantages of stratum ventilation areas are expected (1) single independent system (no background ventilation system is needed); (2) ability to serve a mobile occupants, e.g. the cases of homes and retail shops; and (3) simple and inexpensive design and installation because of no additional ductwork running in the occupied zone. For stationary occupants, personalized ventilation probably performs better, but the aforementioned three factors limit the application of personalized ventilation.

Life cycle assessment (LCA) is a quantitative method to access not only economic and financial viability of investment, but also the environmental impacts of air distribution methods. The stratum ventilation, in comparing with mixing and displacement ventilation are used to estimate the actual energy consumption and greenhouse gas emission during supply and installation, as well as use assessment phase in term of Present value in 5, 10, 15 & 20 service years. The prediction of three ventilation operation performance is based on sensible cooling load estimation and industrial data interpretation, with the scope of energy saving restricted to the life-long electricity saved for the support of system operation, and for reducing the air conditioning system energy use. The greenhouse gas emission is based on the lift cycle inventory in EMSD developer, and supported by another region database for any unavailable information in HKSAR.

This thesis is studying the thermal comfort and cost effectiveness in the environmental chamber by comparing three ventilation methods, including mixing, displacement and stratum ventilation. The performance of six air distribution patterns of the three ventilation methods as shown in the following Figure 1.1 to 1.6 is analyzed by human comfort tests and experimental measurement inside an environmental chamber.

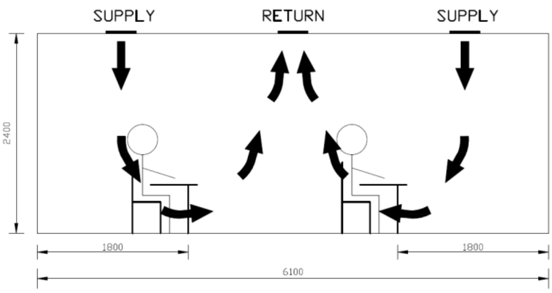


Figure 1.1 Mixing ventilation (MV) with ceiling supply and return

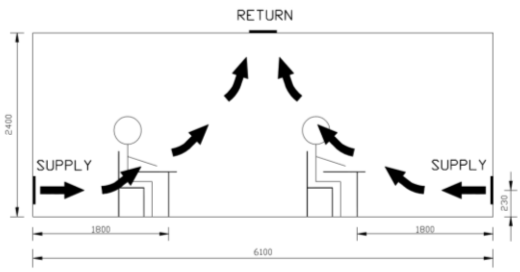


Figure 1.2 Displacement ventilation (DV) with front & rear wall supply at low level and ceiling return

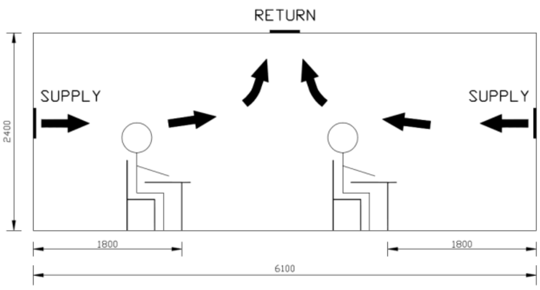


Figure 1.3 Stratum ventilation (SV) with front & rear wall supply at mid-level and ceiling return

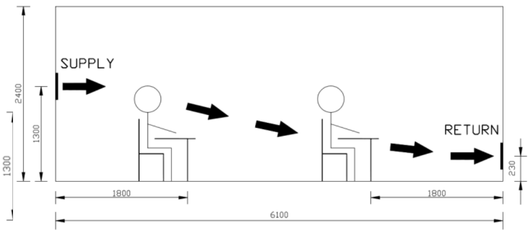


Figure 1.4 Modified-stratum-1 ventilation (SV-1) with rear wall supply at mid-level and front wall

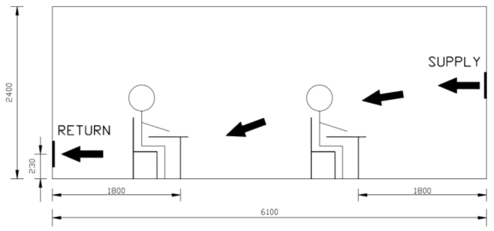


Figure 1.5 Modified-stratum-2 ventilation (SV-2) with front wall supply at mid-level and rear wall return at low level

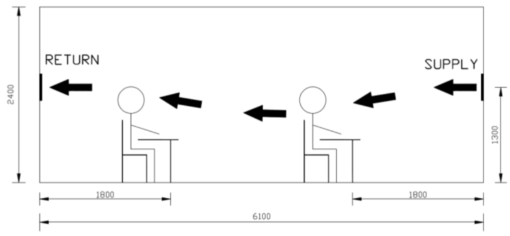


Figure 1.6 Modified-stratum-3 ventilation (SV-3) with front wall supply at mid-level and rear wall return at same level

1.3 Literature review

1.3.1 Thermal comfort

For people living in subtropical regions, their perception of thermal sensation is different from those in the cooler western countries due to acclimatization by Givoni (1998). Nicol (2002) viewed this discrepancy as the result of adaptation and acclimatization. People who are exposed to hot, humid, and lengthy summer, closer to the equator, are getting use to stay in semi-open space with electric fan operation, or natural ventilation. In Brazil, a field survey on natural ventilation buildings by Candido (2009) found that the air velocity should be at least 0.4 m/s for achieving thermal comfort at 26℃ and 0.9 m/s was required if operative temperature reached 30^{o}C. Many subjects were found not only preferring movement, but also indicating preferred air speeds that are either close to or above the 0.8 m/s. Another field survey based on 1520 human subjects in Thailand by Yamtraipat (2005), it found that 26℃, 50 % to 60 % relative humidity and 0.2 m/s air velocity could be used as a comfortable environment condition for the whole country. The neutral temperature for the group of human comfort test with air conditioned acclimatization behavior at home and work was found about 25.4℃. For other group without using air conditioner at home, the neutral temperature of 26.3℃ was found. It was 0.9℃ higher than the air conditioned acclimatization group.

Chung et al. (1990) used a laboratory test of 134 university students in HKSAR, it found that the neutral temperature at clothing level of 0.6 clo under sedentary activities was 24.9℃ and the air speed was about 0.1 m/s. Chow (2010) found that air movement could increase the cooling effect and maintain thermal comfort at elevated temperatures for high heat demand space. This may result in reduced consumption of energy used to cool a building compared with general air conditioning.

Toftum (2004) used ASHRAE field studies in the de Dear (1998) database to examine air movement preference that results are shown in Table 1.1.

Table 1.1 Air movement preferences in the four ASHRAE field studies.

Thermal Sensation	Air Velocity range (m/s)	Percentage of Occupants Preferring on air movement			Sample size
		Want Less	No change	Want More	
Slightly Cool	0-0.15	13.6	46.3	40.1	147
	0.15-0.25	***16.7***	41.7	41.3	48
Neutral	0-0.15	2	46	52	150
	0.15-0.25	2	68.6	29.4	51
Slightly warm	0-0.15	2.7	21.9	***75.4***	73
	0.15-0.25	8.4	33.3	58.3	24

From the data presented in Table 1.1, people with slightly cold perception preferred less air movement and those feeling slightly warm preferred more air movement in between 22.5℃ and 23.5℃ under two local air velocity range. i.e. 16.7% at 0.15-0.25 m/s >

13.6% at 0-0.15 m/s those who felt slightly cool want less air movement; 75.4% at 0 - 0.25 m/s > 13.6% at 0.15-0.25 m/s those who felt slightly want more air movement.

Zhang et al. (2007) stated that office workers with dissatisfaction feeling were very common with too little air movement cited far more than too much air movement based on a database of indoor environmental quality surveys performed in over 200 buildings. In this survey of thermal comfort, near half of the participants preferred more air movement and only 4% from them expressed the choice of less air movement while the velocity of air movement in workspaces was higher than 0.2 m/s, which is the de facto draught limit in both the current ASHRAE as well as ISO thermal environment standards. With an air velocity of 1 m/s in breathing zone at warm temperature, the air movement was interpreted as acceptable. In contrast, most of the participants could not accept still air in warm environments (Zhang et al. 2008).

Some laboratory and field studies indicated that effective and high enough air movement took a very vital role for compensating for warm temperatures in generating comfort to general public (Tanabe 1989; Fountain et al. 1994; Toftum 1997, Nicol 2004 & Li 2005). People are prone to request for more air movement when they feel neutral or above on the thermal sensation scale. Human comfort study conducted by forty subjects, Toftum (1997) found that airflow direction has an impact on perceived discomfort due to draught and room air temperature. This study involved 20 women and 20 men, were exposed to airflows from five different directions: horizontally towards the front, the back, and the left side and vertically upwards and downwards. The subjects were exposed to stepwise increased air velocities ranging from less than

0.10 m/s to 0.40 m/s at three room temperature levels 20, 23 and 26℃. Quantitatively, the effect depended on the air temperature. At 20℃ and 23℃, most subjects perceived discomfort when exposed to air movements from below, whereas at 26℃ most subjects perceived discomfort at air movements from above. Side directions were considerably less sensitive to air movement but it might be the case for the one at the front direction during warm conditions.

Arens et al. (2009) found that air movement in warm environment could probably bring away certain amount of body heat which enhanced the perception of improved air quality resulting in better thermal comfort. Therefore, instead of providing energy consuming air conditioning facility, more energy-efficient and conserving air movement at indoor environment is good enough to generate thermal comfort to general public.

Zhang et al. (2007) suggested that by broadening the current draught limit for neutral-to-warm conditions would encourage energy saving which are now only restricted to individual manipulated air movement devices. Although it is ideal for general public to control their personal desirable air movement, the practice for achieving this is still very limited and restricted. It is because of the common ventilation system adopted in HKSAR is mixing ventilation. These researches proposed that there should be more combination for the range of temperatures and air velocities for the device in which large areas could be involved without inducing possible draught risk for the occupants.

Some experimental and numerical investigations have been conducted to evaluate the performance of stratum ventilation. Tian et al. (2011) comprehensively measured the distributions of air speed, temperature and carbon dioxide concentrations in a stratum ventilated office. The results showed that stratum ventilation provided satisfactory thermal comfort level quantified by *PMV* and *PPD.* The ventilation effectiveness was close to 1.5 and showing that stratum ventilation was efficient because for perfect mixing ventilation is equal to 1.

Thermal comfort is one major indicator to realise the actual performance of air conditioning system. The common aim is to find out the acceptable indoor conditions for occupancy in different regions for design and control of the air conditioning mode in buildings. Recently, with the rising concern on worldwide climate change, more attention is paid to the study of any possible air distribution system to minimise the energy consumption in order to reduce carbon footprint. Adopting higher room temperature and air speed are explored to achieve the most desired thermal comfort level.

A study in HKSAR found that the neutral temperature for air-conditioned offices under mixing ventilation method in subtropical climates were 23.6^{o}C in summer (Mui et al. 2007). The result was evaluated from 422 occupants' responses towards the perceiving thermal environment in 61 air-conditioned office buildings in HKSAR. In other study (Chung et al. 1990), the neutral temperature of young HKSAR Chinese was found to be 24.9℃ which was based on 134-college-age Chinese subjects wearing 0.6 clo standard clothing, under 0.1 m/s of mean air speed and sedentary activity.

Fong et al. (2010) developed the thermal sensation model based on the regression analysis of the vote results of ASHRAE 7-point scale from 203 human subjects aged 19-

21 participated. The overall thermal sensation as a function of air temperature, speed and humidity ratio is expressed in the following Equation 1.1. The mean radiant temperature, activity level and clothing insulation are kept as constant during thermal comfort survey.

$$Ts_o = 0.1818\, t_c - 0.426\, u + 34.02\, w - 4.702 \qquad (1.1)$$

Where

Ts_o : Overall thermal sensation

t_c: air room dry-bulb temperature (^oC)

u: air speed (m/s)

w: humidity ratio (kg/kg)

From Equation (1.1), it appears that all the three parameters including air temperature, air speed and humidity ratio, has a similar effect on the thermal sensation, but thermal sensation is not sensitive to the humidity ratio. When thermal sensations are determined based on the humidity ratios corresponding to 50%, 65% and 80% RH, the average deviation of thermal sensation between 65% and 80%RH were found to be only 4.7% compared to that at 50%RH with the same temperature and air speed. As result, only temperature and air velocity under acceptable range of 50-80% RH are going to be used in the ongoing analysis of my study. This finding is also in line with the results of Givoni et al. (2005) with subjects from Indonesia, Singapore, Thailand and Japan.

With air speed at 0.1 m/s to 0.2 m/s, measured at 0.6m above floor level, 300 colleague students with 0.55 clo standard clothing, metabolic rate 1 met, the neutral temperature of 25.4^oC for sedentary working environment was found by Chow et al. (2009). A laboratory and field study found that higher air movement can compensate for warm temperatures in terms of making people comfortable in summer conditions by Srivajana (2003). This finding was also in line with the recently revised ANSI/ASHRAE Standard 55-2010. It allows a wider range of operative temperatures to be considered as acceptable thermal conditions with a higher air speed. For instant, an air speed of 1.2 m/s under 0.5 clo, 1.1 met, 31℃ is still acceptable and within the comfort zone as shown in ANSI/ASHRAE Standard 55-2010. It is desirable to achieve acceptable thermal sensation with the minimum use of energy by increasing air speed, air movement, adjusting supply air positions in the occupied zone from a building sustainability point of view to provide a possible way in ventilation system design.

To balance thermal comfort and energy performance, the neutral thermal sensation of HKSAR people at elevated room temperature of various systems was determined by a thermal comfort survey (Fong et al. 2010). The stratum ventilation system may be able to provide temperature and air movement which may be comparably higher but within the range of ASHRAE 55-2010. Thus, thermal comfort survey study and energy performance analysis for this stratum ventilation in HKSAR is therefore justified. Due to lack of comprehensive experimental data to estimate the neutral temperature in subtropical regions, human comfort test and field measurements have been carried out to identify the actual benefits of stratum ventilation. Temperatures and air velocities were measured at specific locations in the chamber, during morning, noon and afternoon hours of typical summer days.

Thus, human subject test is required to determine the effect of thermal comfort condition by different combination of temperature and air flow rate. The ASHRAE Standard 55-2010 indicate that allow increased air movement to broadly offset the need to cool the air in warm conditions. The performances, especially in thermal comfort, of a non-conventional air distribution system are not deducible based on existing literature. Lin et al. (2005, 2009) developed a new air distribution method named stratum ventilation which can accommodate a higher room temperature. Experimental and computational results from test chamber at Xi'an Jiaotong University (Tian et al. 2008, 2011) showed that with properly designed supplied air velocity and volume; this air distribution method can maintain better thermal comfort and ventilation efficiency. Thermal comfort indices and ventilation effectiveness were found to satisfy the requirement of ISO7730, CR 1752-1988 and ASHRAE 55-2010 and fully matched with the previous findings from Toftum (1997) and Mayer E (1992).

1.3.2 Computational fluid dynamic model

Computational fluid dynamics (CFD) is a valuable tool in the analysis and design of ventilation systems. The application of CFD for indoor spaces can tackle the complexity of the flows involved and of the interaction between the various modes of heat transfer with concerning the accuracy of the results obtained. More comparison between CFD results and experimental results would answer some of these concerns. There are many available turbulence models. Rohdin and Moshfegh (2007) presented a comparison between three eddy-viscosity turbulence models, namely standard k-ε, RNG k-ε and realizable k-ε, used for predictions of the flow pattern and temperature distribution in the large industrial facility.

Tian et al. (2009) employed a numerical method validated by experimental data to predict the diffusion of gaseous contaminants in a stratum ventilated space (Tian et al. 2010). The results were compared with displacement ventilation and found that the flow pattern formed by stratum ventilation was able to provide better inhaled air quality in term of carbon dioxide concentration. Tian et al. (2009) also investigated indoor aerosol particle dispersion under stratum ventilation and displacement ventilation by numerical analysis. Compared with displacement ventilation, stratum ventilation exhibited better ventilation performance and both the particle concentrations of the entire room and of the breathing zone become lower. The anti-airborne infection performance of stratum ventilation was evaluated by Lin et al. (2012). The results of simulation demonstrated that the particle concentration in breathing zone under stratum ventilation was significantly less than displacement ventilation, which implies that the risk of pathogen inhalation under stratum ventilation is lower.

The objective of this work is to apply a Re-Normalization Group (RNG) k-ε model, which is a reliable model among the eddy-viscosity models tested (Chen 1995 & Tian 2010). Firstly, Tian et al. (2009) using a numerical method in Xi'an Jiaotong University to investigate an indoor air quality and thermal comfort of an office room with stratum ventilation. The experimental results for an office with the size of 3.9m x 2.6m x 2.9m (length x width x height) served by stratum ventilation with comprehensive, long-time measurements. It was shown that stratum ventilation could provide good indoor air quality in the breathing zone and to achieve good thermal comfort measured by *PMV* and *PPD* with reasonable design. Secondly, Lin et al. (2010) developed an RNG k–ε model based on the commercial program of FLUENT to perform CFD analysis. This model has been experimentally validated for stratum ventilation and investigated the diffusion of carbon dioxide exhausted from the subjects and achievable thermal comfort level under stratum ventilation. The results demonstrated the flow pattern formed by stratum ventilation could provide good IAQ in the breathing zone. The particle concentrations of the entire room and the breathing zone under stratum ventilation are less than those under displacement ventilation. It implied that the risk of inhaling particles, particularly in the breathing zone, is smaller under stratum ventilation than under displacement ventilation by adopting the numerical method validated by their experimental data. In comparing with field measurement and predicted result by RNG k-ε model, it was found these were concurrent. The contaminants transport and air flow could be determined by computationally solving a set of conservation equations describing the flow, contaminants and air age in concern system. Generally, the airflow is a complicated convection driven by momentum and buoyancy producing three-dimensional turbulent flow.

Thus, Re-Normalization Group (RNG) k-ε model under FLUENT program, which is a reliable model among the eddy-viscosity models tested (Chen 1995 & Tian 2010), has been used to assist the environmental chamber setup. It can realize a workability of conducting a human subject tests within this chamber serving by different ventilation modes. The preliminary idea from engineering design, such as the location of each supply and return terminal, setting out of seating layout, as well as the range of supply air flow for each ventilation mode should be reconfirmed prior to carry out the procurement and construction works.

1.3.3 Cost effectiveness

This study utilizes a recently developed Life cycle assessment and Life cycle costing tool for commercial building developments in Hong Kong special administration region (HKSAR) especially in user manual, Electrical and mechanical service department (EMSD), HKSAR government publication and additional inventory data base from ECOINVENT via "eBalance" platform. Life cycle costing (LCC) has been widely adopted in the assessment of building systems. LCC has taken into account of initial, operation, maintenance and salvage cost for the related service and life time of building system. The study utilizes the producer price version of the Ecoinvent model as supplementary information to support and rectify the existing EMSD database. Ecoinvent Center holds the world's leading database with consistent, transparent and up-to-date Life cycle inventory data, with several thousands of LCI datasets in the various areas, such as agriculture, energy supply, transport, biofuels and biomaterials, construction materials, etc. "SimaPro" and "ebalance" come with the full Ecoinvent database covering more than 4000 processes and become one of the most comprehensive international inventories. The inventory database of different countries and regions with respect to defined system boundary are including the Ecoinvent, ELCD, CLCD, ICE, and LCI. Figure 1.7 shows the Swiss organizations that joined forces to create the Ecoinvent database.

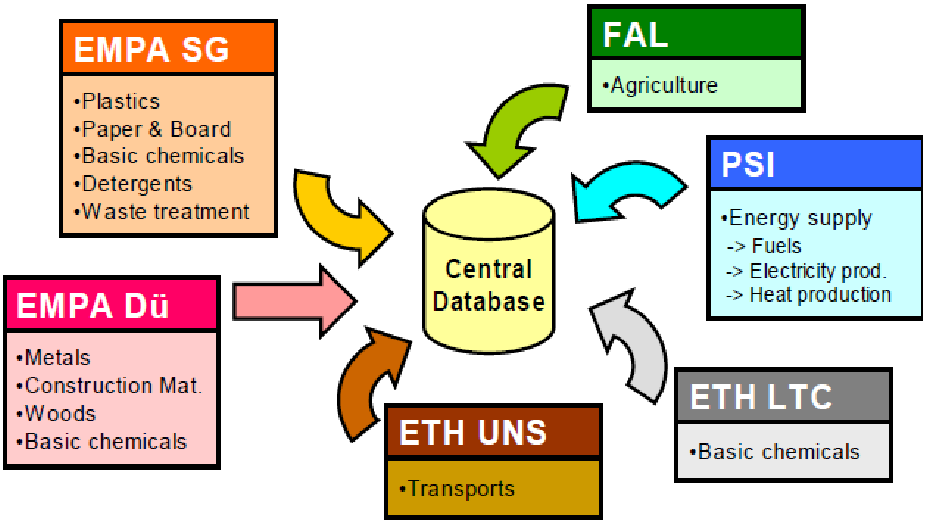


Figure 1.7 Ecoinvent database

1.3.4 Recent study of stratum ventilation

In recent five-year study, the medium size of stratum ventilated room is used to investigate by some researchers. The related matters with my study of thermal comfort and cost effectiveness for stratum ventilation are highlighted. This will form an important reference for my study in term of thermal comfort and cost effectiveness.

1.3.4.1Energy performance of three ventilation system

Lin et al. (2011, 2013) studied the energy performance of various ventilation methods in typical configurations of an office, a classroom and a retail shop. The year-round energy consumption of stratum ventilation was found to be substantially lower than mixing ventilation and displacement ventilation.

Lee et al. (2013) concluded that the energy saving potential of stratum ventilation derived from reduction of ventilation load, transmission loads and increased *COP* of chillers. The year-round energy saving in classroom was found to be substantial at 26.6% and 41.1% at least when compared with displacement ventilation and mixing ventilation, respectively. This study is also focused on how much energy can be saved due to the neutral temperature and supported by experimental results for the installed mixing, displacement, and stratum ventilation methods inside an environmental chamber. This study analyses in terms of energy consumption and present worth according to this year-round approach.

1.3.4.1 Design Parameter of three ventilation systems

The findings of mixing, displacement and stratum ventilation systems in terms of work principle, supply air velocity and temperature, energy saving and efficiency of heat removal are tabulated in Table 1.2.

Table 1.2 Comparison of three ventilation systems

Ventilation System	Mixing Ventilation	Displacement Ventilation	Stratum Ventilation
Work Principle	Treat the room air by mixing the ceiling supplied air together with pre-treated outdoor air within the entire room.	Displace the supplied air from floor level	Directly supply the air into the occupant's breathing zone.
Supply air velocity	Around 1.5 m/s Awbi (2011)	Less than 0.5 m/s Awbi (2011)	1.0 to 2.0 m/s Lin Z et al. (2011)
Supply air temperature (Cooling Mode)	14 to 22 ℃ Awbi (2008)	18 to 24℃ Lee et al. (2009); Causone et al. (2010)	19.5 to 23℃ Lin et al. (2011)
Energy Saving in Classroom	Reference case	20-34% Awbi (2000)	44-75% in comparing with MV; 25-56% in comparing with DV Lin Z (2011); Lee (2013)
Efficiency of heat removal [#1]	0.99 Karimipanah (2008)	1.21 Karimipanah (2008)	1.5 Tian (2011)
[#1]Efficiency of heat removal, $E_{Heat\ removal} = \frac{T_{exhaust\ air} - T_{supply\ air}}{T_{occupied\ zone} - T_{supply\ air}}$ where T is temperature.			

1.4 Research objectives

The aims and objectives are:

a. To develop a full-scale environmental chamber, including various air distribution methods, all measuring devices, and monitoring & control system to achieve the following objectives.
b. To find out a neutral temperature for mixing, displacement, stratum air distribution methods supporting by human comfort test and experimental data;
c. To find out an uniformity of measured data in stratum ventilation for supporting the significant level of collected data from human comfort test, and results of the neutral temperature of each ventilation method
d. To differentiate energy conservation and cost effectiveness of mixing, displacement, stratum air distribution methods based on current market conditions.

1.5 Research methodology

In order to achieve the objectives in this research work, the following methodology has been adopted:

a. To conduct the literature survey of human comfort tests, with the focus on air movement preferences, and energy cost on various air distribution strategies.

b. To build up the real scale chamber with realistic boundary conditions by FLUENT program for assisting the finalization of each installation detail of real environmental chamber together with mixing, displacement and stratum ventilation for conducting the human comfort test and field measurement.

c. To conduct the surveys on satisfaction level of thermal comfort and monitor all boundary conditions and operating parameters in mixing, displacement and stratum ventilation.

d. To develop the plan for human comfort tests and field measurement related to various air distribution methods. Thermal comfort questionnaires, field measurement devices, recruitment of sufficient subjects are conducted. The detail of collection for this human comfort test is based on ethical requirement of Faculty Human Research Ethical Committee, DMU.

e. To analysis all collected data from the human comfort tests under the testing boundary conditions and operating parameters in mixing, displacement and stratum ventilation.

f. To carry out the laboratory field studies and human comfort test to explore the thermal comfort and cost effectiveness for this ventilation system applied in subtropical HKSAR region.

g. To evaluate the air distribution methods by analysing the aforesaid performances to produce optimized design of stratum ventilation.

h. To compare the cost effectiveness of stratum ventilation based on mixing, displacement ventilation.

i. To analyses all the empirical & experimental data, as well as systematic record of human comfort test to form as basis of preparing the thesis writing.

The following facilities and resources are used for my research work:

- Unique environment chamber with mixing, displacement and stratum systems and full set of measurement instrumentation for thermal comfort parameters;
- Thermal Comfort Data Logger, INNOVA 1221 model built up modularly with input modules with several number of transducer sockets;
- Temperature, air velocity and humidity transducers connected to the Data Logger;
- HP Z800 Workstation (2 Intel Xeon X5675 3.06 12MB/1333 6C CPUs, HP 24GB (6x4GB) DDR3-1333 ECC 1-CPU RAM);
- Computation Fluid Dynamics (CFD) simulation program "FLUENT".

1.6 Organization of thesis

This thesis is organized into six chapters, each chapter dealing with a particular aspect of the research on "A Study of Thermal Comfort and Cost Effectiveness of Stratum Ventilation".

Chapter 1 introduces the growing threat of climate change; several governments in East Asia have made proactive moves by issuing new energy ordinance and guideline on temperature control in air-conditioned premises. This chapter provides a potential ventilation method to minimise the carbon emission and satisfy with thermal comfort perception. Six ventilation modes are used to assess the thermal comfort, energy performance, as well as better cost effectiveness measures in in preset condition in the environment chamber. The contents also indicate the literature review on thermal comfort, cost effectiveness aspects for three ventilation methods. The end of this chapter describes how to apply the research methodology to achieve the objectives.

Chapter 2 is mainly focused on the physical environmental chamber setup, assisted by the computer simulation result to establish overall systematic air distribution methods and all system control and monitoring, measuring devices to collect all necessary data for human comfort test and field measurement.

Chapter 3 addresses the thermal comfort analysis aspects by means of collected data analysis from all human comfort tests in different setting of six air distribution methods including mixing, displacement and stratum ventilations. Thermal neutral temperature and energy consumption of each ventilation method are found in the environmental chamber. Modified-3-stratum, which is the lowest energy consumption ventilation method, is selected to carry out the further study.

Chapter 4 concentrates on using the modified-stratum-3 for further evaluation and then, explores the uniformity of thermal environments in the occupied zone by three series tests. For most cases studied, the values of air distribution performance index (ADPI) derived from objective experimental measured data and the standard deviations of human subjectively thermal sensation on ASHRAE 7-point scale test are assessed. It provides scientific basis for using existing thermal comfort parameters to describe stratum ventilation for air distribution design.

Chapter 5 focuses on Life cycle assessment (LCA) for these case studies of the full-scale mixing, displacement and stratum ventilation systems installed inside the environmental chamber. The simple energy and cost payback period of three ventilation methods are estimated. LCA methodology is developed to evaluate which technical energy design solution is more sustainable in the long run.

Finally, **Chapter 6** is the conclusion and the recommendations on the future works. The analysis methodology in term of thermal comfort, energy conservation, as well as economic and environmental assessment is applicable for another similar ventilation systems design.

CHAPTER 2: DESIGN OF THE ENVIRONMENTAL CHAMBER

2.1 Experimental chamber setup

This research will deal with optimal thermal environments for human beings, methods of evaluating a neutral temperature, and the principles for the establishment of a detailed thermal analysis, energy use and cost effectiveness for a given thermal environment served by various air distribution methods in subtropical region of HKSAR. Thus, a full-scale environmental chamber is indispensable for addressing the objectives as mentioned in Chapter One. Before any physical works, computational fluid dynamic (CFD) simulation program is used to assist the design of all necessary components inside the chamber.

2.1.1 Air distribution strategies

Differentiation of various air distribution strategies under preset boundary condition is a classical and efficient investigation method to determine the actual benefit of stratum ventilation. Air distribution strategy is considered taking account of what kind of special ventilation mode is to be used together with type, number, and setting-out of each terminal to be adopted for serving an air conditioned area. Boundary condition is often defined as the collective whole of all the physical properties in a room which influence a person by supply air speed, air flow rate, and temperature setting to match with cooling intensity requirement, architectural layout, room dimension, seating position. Due to the environmental chamber situated along the core of air-conditioned building, not close to any external wall, only fresh air ventilation load will be affected by the outdoor weather condition, and specifications of heat transfer from external heat transfer can be drawn up independent of the thermal environment.

The above considerations are identified as input parameters inside the environmental chamber to affect the thermal comfort achievement in respect to energy conservation; those are illustrated in Figure 2.1. Figure 2.2 illustrates the possible outcomes from the input parameters inside the environmental chamber. Neutral temperature and acceptable percentage from human comfort tests have been studied with respect to six ventilation modes. The experimental chamber was placed in operation in 2010. Geometry of the chamber and its adjoining facilities is shown in Figure 2.3. The chamber is 6.1m wide by 8.8m long with a ceiling height of 2.4 m. In the chamber all relevant combinations of air temperature, air humidity, mean radiant temperature and air velocity can be produced and accurately controlled.

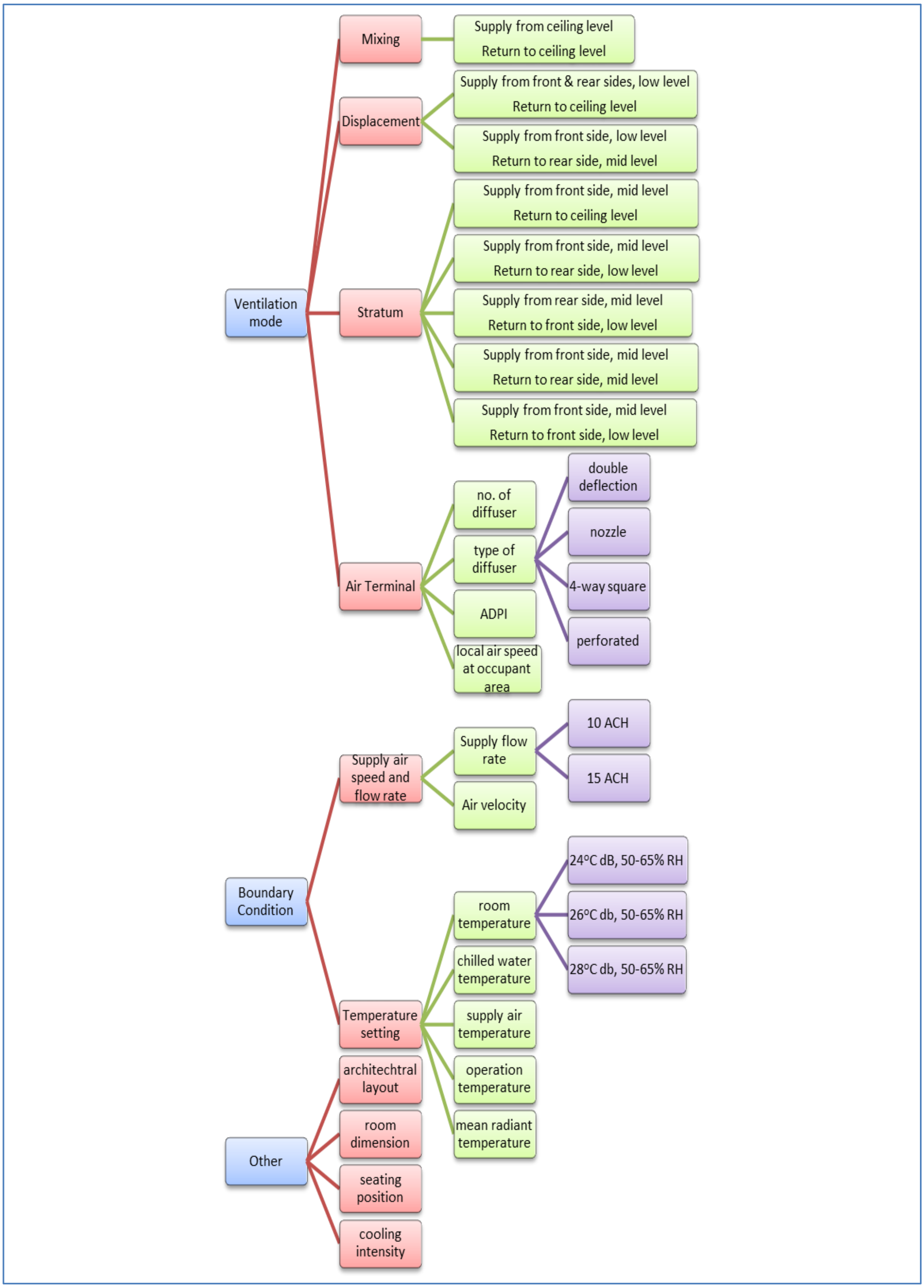


Figure 2.1 Input parameters inside environmental chamber

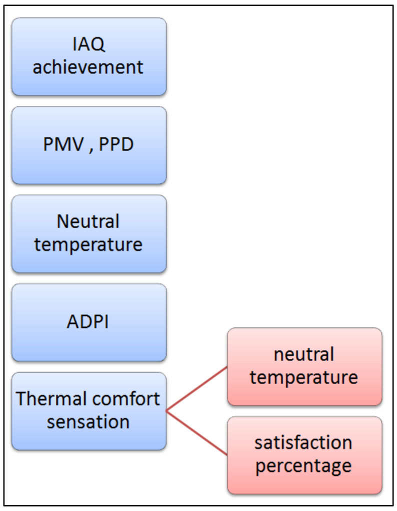
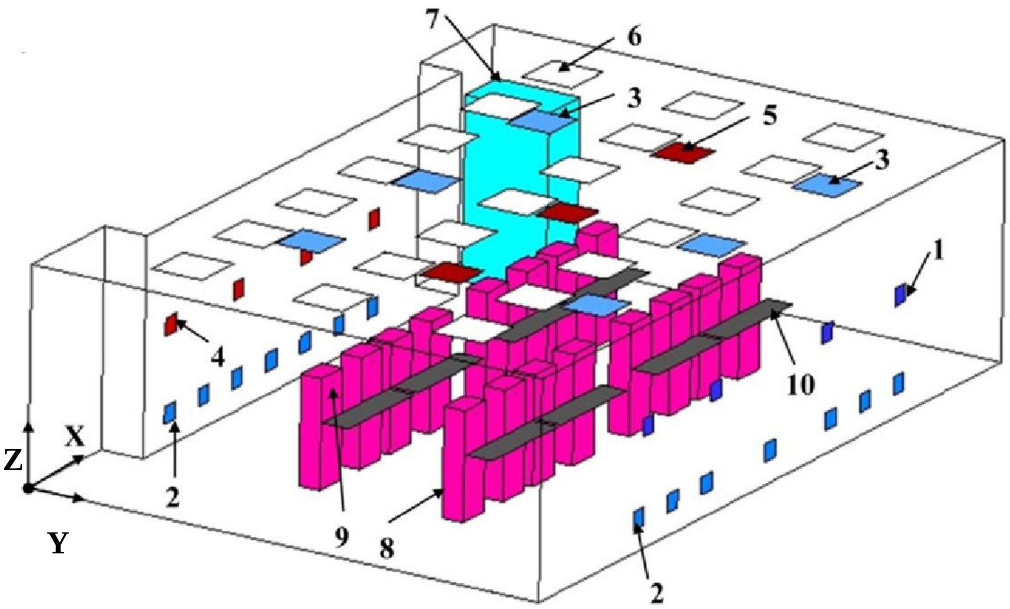


Figure 2.2 Expected outcome from environmental chamber



1-Inlets for SV, SV-1 & SV-3 or Oultets for SV-2; **2**-Inlets for DV; **3**-Inlets for MV;
4-Outles for SV,SV-1 & SV-3 or Inlets for SV-2; **5**-Outlets for DV & MV; **6**-Lighting panels;
7-BMS workstation; **8**-Seated occupant; **9**-occupant's mouth; 10-desk.

Figure 2.3 Geometry of Environmental Chamber

2.1.2 Computational fluid dynamic model

The Computation Fluid Dynamics (CFD) simulation program "FLUENT" has been used to develop the mathematical models. This CFD program with the RNG k- ε model of turbulence has been firstly used to predict the air flow pattern, the temperature profiles, for an environmental chamber under stratum ventilation.

The CFD model was constructed and validated using experimental measurement as shown in Figure 2.4. It is for a ventilated enclosure, the size of environmental chamber is similar to typical classroom in HKSAR, equipped with a conventional mixing, displacement and stratum ventilation systems.

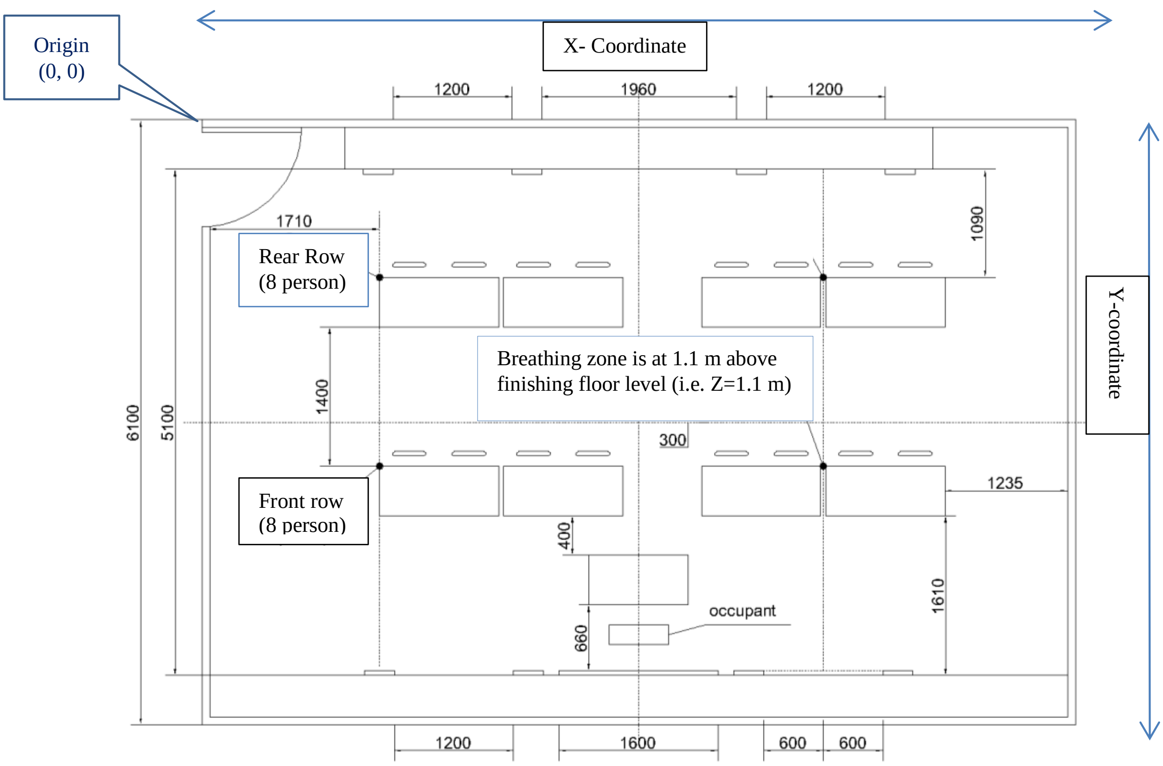


Figure 2.4 Layout of simulation model

The boundary conditions of the above model imposed on the related partial differential equations are implemented using RNG k-ε model in order to predict the flow pattern.

The terminal is defined as an opening with a uniform velocity. Outlet boundary conditions are set as the Neumann boundary condition, i.e., mass flow boundaries are specified to ensure that the mass flow rate out of the domain is the same as the mass flow rate into the flow domain (Zhao 2004). The walls of heat sources are defined as constant heat flux. Since the RNG k-ε model is valid for high Reynolds number turbulent flow, wall functions are needed for near wall region where flow Reynolds number is low. The present investigation uses the standard wall function to describe the turbulent flow properties in the near wall region. More details are given in Launder and Spalding (1974). In order to guarantee the validity of wall function, denser grids are adopted in the near wall region, which can be checked by the value of wall-bounded turbulent flows y^+ with equation (2.1) in the region. The wall y+ is a non-dimensional distance similar to local Reynolds number, often used in CFD to describe how coarse or fine a mesh is for a particular flow. It is the ratio between the turbulent and laminar influences in a cell. Where: Density of fluid, kg/m^3; y: Vertical distance normal to wall direction, m; u_τ: Shear stress velocity, m/s; υ : Kinematic viscosity of fluid, (m^2/s)

$$y^+ = \frac{\rho y u_\tau}{\upsilon} \qquad \text{Eq. (2.1)}$$

For the following simulation, the maximum value of y^+ is around 51 which can satisfies the acceptable range of accuracy level to be studied by Salim.M.Salim and S.C. Cheah (2009) in Fluent. The heat resource value of boundary conditions for the front lamp,

lamp, occupant, workstation, and wall are listed on Table 2.1. The heat flux is calculated by equation of q=Q/A (W/m^2), where Q is heat generated W; A is surface area, m^2. The location of each source is shown in Figure 2.3.

Table 2.1 Heat flux of each heat source in this model

Item	Heat Flux (W/m^2)
Front lamp	183.1
Lamp	147.7
Occupant	42.2
Workstation	43.6
Wall	0

The number of grid in the simulation mode is 579 612, which has been checked to guarantee grid independence. The converged residuals for continuity equation and u_j are 10^{-3}, for other scalars such as T, k, ε, C and τ, the converged residuals are 10^{-5}.

Comparison with various ventilation modes shows that the stratum ventilation can maintain thermal comfort and indoor air quality based on identical fresh airflow rate and air circulation rate. Example of providing a convenient base for energy performance comparison of the various cases is based on the related heat radiant rate by internal elements.

2.1.3 Air movement control

The movement of the air can be predicted from the solution of the discretized mass, momentum and energy conservation equations. The transport general equations in tensor notation are expressed as follows:

$$\frac{\partial}{\partial t}(\rho\phi)+\frac{\partial}{\partial x_j}(\rho u_j\phi)=\frac{\partial}{\partial x_j}\left(\Gamma_\phi\frac{\partial\phi}{x_j}\right)+S_\phi$$

Eq. (2.2)

where ϕ represents each of the three velocity components *u, v, w* in momentum conservation equation; the kinetic energy of turbulence k, the dissipation rate of the kinetic energy of turbulence ε, air enthalpy h in energy conservation equation; and ϕ =1 in mass conservation equation. The volume concentration of carbon dioxide *c*, and air age τ are user defined transported scalars. Γ_ϕ is the effective diffusion coefficient and S_ϕ is the source term of the general equation. The source term and the diffusion coefficient corresponding to each variable ϕ solved for in this study are given in Table 2.2. Yoshihide and Tominaga (2007) conducted the CFD analysis of flow and dispersion around an isolated cubic building using RNG k–e model with various Sc_t. The results show smaller value which is closer to the data collected from experiment. The equations are discretized into algebraic equations by the finite volume method (FVM). The second-order upwind scheme is used to variables such as pressure, momentum, energy, turbulence kinetic energy, turbulence dissipation rate, carbon dioxide concentration and air age which are realised by user defined scalar. The Boussinesq assumption is employed to consider the buoyancy effect. Moreover, the discrete ordinates (DO) radiation model is used to take radiation of walls into consideration (Abanto et al. 2004).

Table 2.2 Source and diffusion coefficient for each parameter

Variables	ϕ	Γ_ϕ	S_ϕ
Mass conservation equation	1	0	0
Momentum equation	u_j	$\mu+\mu_t$	$-\frac{\partial p}{\partial x_j}-\rho\beta g_j(T-T_0)$
Energy equation	k	$(\mu+\mu_t)/\delta_k$	$G-\rho\varepsilon+G_B$
Turbulent Kinetic energy equation	ε	$(\mu+\mu_t)/\varepsilon_\varepsilon$	$(C_{\varepsilon 1}G-C_{\varepsilon 2}\rho\varepsilon+C_{\varepsilon 3}G_B)\varepsilon/k+$
Turbulent dissipation equation	T	$\mu/\delta_1+\mu_t/\delta_t$	S_T
Component equation	C	μ_t/Sc_t	S_C
Air age	τ	μ_t/Sc_t	ρ

where

μ is laminar viscosity; $\mu_t=\rho C_\mu \frac{k^2}{\varepsilon}$ is turbulent viscosity

$G=\mu_t\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)$ is the turbulent production

$G_B=-g_i\beta\frac{\mu_t}{\Pr_t}\frac{\partial T}{\partial x_i}$ is the turbulent production due to buoyancy

$R=\frac{C_\mu\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}\frac{\varepsilon^2}{k}$ is the source term from renormalization

$\eta=S\frac{k}{\varepsilon}$, $S=(2S_{i,j}S_{i,j})^{1/2}$, $S_{i,j}=\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)$

$C_\mu=0.0845$, $C_{\varepsilon 1}=1.42$, $C_{\varepsilon 2}=1.68$, $C_{\varepsilon 3}=1.68$ are the model constants

$\delta_k=0.7194$, $\delta_\varepsilon=0.7194$, $\delta_1=0.7$, $\delta_t=0.9$, $Pr_t=0.7$ and $Sc_t=0.7$ are used in the simulation as default values.

Remark:"Pr_t- Prandtl number" represents the analogue of flow and heat transfer; and "Sc_t = Schmidt number "represents the analogue of flow and mass transfer.

FLUENT using the discrete ordinates (DO) radiation model solves the radiative transfer equation (RTE) for a finite number of discrete solid angles, each associated with a vector direction fixed in the global Cartesian system. The fineness of the angular discretization can be controlled. The effect of a discrete second phase of particulates on radiation can be allowed in DO model. In this case, FLUENT will neglect all other sources of scattering in the gas phase.

Stratum ventilation is expected to show a strong nature of bifurcation because the buoyancy and the momentum force are both of the same orders of magnitude due to the horizontal supply air path. The swings of the supply air jets would have significant impacts on air age, thermal comfort, especially if the swing frequency is of the same order of magnitude as the turbulence pulsation.

2.1.4 Velocity and temperature profile

This CFD analysis was performed on the subjects of thermal comfort, ventilation effectiveness and indoor quality in a realistic ventilated enclosure. The issue of occupant modeling was approached by simply assuming uniform heat distribution over the entire body and by using blocks as segment of the body. A single block of rectangular shape is used to model the whole body instead of multi-node configuration to simplify the case of occupant configuration. The whole-body thermal comfort equation is function of the air velocity, air temperature, metabolic rate, clothing value, relative humidity and mean radiant temperature. Any significant change in any of these parameters influences the correct prediction of the overall thermal comfort (Fanger 1970).

All the variants of computational fluid dynamics (CFD) methods are based on the governing equations for fluid flow which have been known for over century. The fluid governing equations are derived from the laws of conservation of mass, momentum and energy. The indoor air quality as well as thermal comfort can be determined by computationally solving a set of conservation equations predicting the temperature & air velocity profile, *PPD*, *PMV*, carbon dioxide and air age in a designed system. Numerical solution of these conservation equations provides a reference and optional idea prior to advocate the reality work in system design. Generally, the flow of air is a mixed convection driven by momentum and buoyancy which is three-dimensional turbulent flow. There are many turbulence models available. The standard k-e model is probably most widely utilized in engineering calculations due to its relative simplicity. Eight different types of turbulence models were investigated by Chen (1995) to determine the most appropriate model for indoor air flow computations. He concluded that the Re-Normalization Group (RNG) k–ε model was the most accurate model among the eddy-viscosity models tested. Four turbulence models (1) k-epsilon-standard (k-ε STD), (2) k-ε-renormalization group (k-εRNG), (3) k-ε-realizable models and (4) k-ω-shear stress transport (k—SST) models are available in a commercial CFD solver FLUENT to be used in studying for the assessment of indoor thermal environment. For this study, RNG k–ε model was used to compare with the experimental data.

Due to site constrain, the spatial arrangement shall not deviate with the exiting architectural layout plan. This environmental chamber shall be used as classroom for teaching purpose during assigned school day. Before actual installation of the environment chamber, the pre-determined items, such as setting out and number of inlets and outlet terminates, as well as the furniture layout shall be simulated and realized the feasibility to conduct thermal comfort survey. The construction concept of Environmental chamber is shown in Figure A4, section A1.3 Ventilation systems, Appendix A. Four studies cases together with same supply air flow of 10 air change per hour and various supply air temperatures are tabulated in Table 2.3. The major purposes of this simulation is visualized the temperature distribution layout and airflow pattern at different level based on the conceptual construction detail and pre-set room condition for further thermal comfort survey and experimental measurement.

Table 2.3 Varies cases for CFD simulation

Case	Description of ventilation method and boundary condition
1	Stratum ventilation with 4 nos. 195 mm x 195 mm front wall supply and 4 nos. 195 mm x 195 mm rear wall return at middle level (Supply temperature: 21℃ (or 294K) and Supply air velocity: 2.48 m/s at 10ACH)
2	Conventional mixing ventilation with 6 nos. 600 mm x600 mm 4-way ceiling supply and 3 nos. 600 mm x 600 mm louver return (Supply temperature: 19.5℃ (or 292.5K); Supply velocity: 0.63 m/s)
3	Conventional mixing ventilation with 6 nos. 500 mm x400 mm 4-way ceiling supply and 3 nos. 500 mm x400 mm louver return (Supply temperature: 19.5℃ (or 292.5K); Supply velocity: 0.32 m/s)
4	Stratum ventilation with 4 nos. 195 mm x 195 mm front wall supply and 4 nos. x195 mm x 195 mm rear wall return at middle level (Supply temperature: 19.5℃ (or 292.5K) and Supply velocity: 2.48 m/s)

2.1.5 Computational fluid dynamic analysis

In Figures 2.5 & 2.6 shows the flow pattern and temperature gradient distribution, which computed with the CFD program under 21℃ (or 294 K) supply air temperature and 10 air change per hour. X-coordinate, Y-coordinate and origin of this chamber for each simulation case is illustrated in Figure 2.4. This case 1 has higher supply air temperature 21℃ (or 294 K) than the other three cases of 19.5℃(or 292.5 K). Due to buoyancy, the horizontally supplied cold air from the diffuser dips slightly, because the existence of occupant and table, several eddy are formed. In the region close to the occupant, the air flows upwardly due to the presence of the occupant. The supply airflows through the breathing zone were computed for this warm-neutral temperature. It found the airflow has gradually dips because of temperature difference at other zone. Air plumes occurred around the occupants due to convention. From this simulation result, the breathing zone is between 0.8 m to 1.4 m above finishing floor level where the air velocity is prevailingly horizontal and higher than that of the other zone.

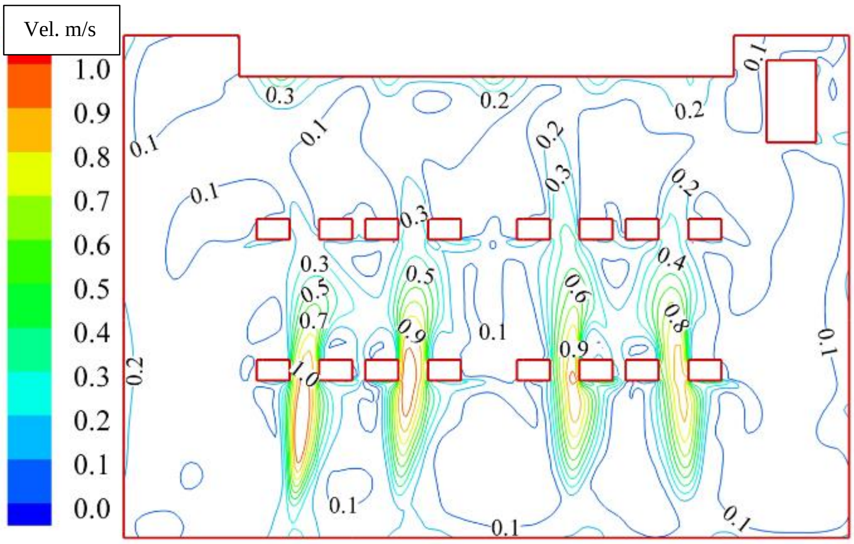


Figure 2.5 Air velocity distribution layout (m/s) at Z =1.1 m
(Case 1: Supply air temperature = 21 ℃ (or 294K))

Figure 2.5 shows the range of air velocity from 1.0 m/s to 0.3 m/s at the breathing zone of human subjects at front and rear row. The distance of 1860 and 3570 mm in between the supply air terminals and two-row human subjects is considered acceptable to adopt in human comfort tests and local discomfort due to draft is not significantly affecting the thermal comfort result.

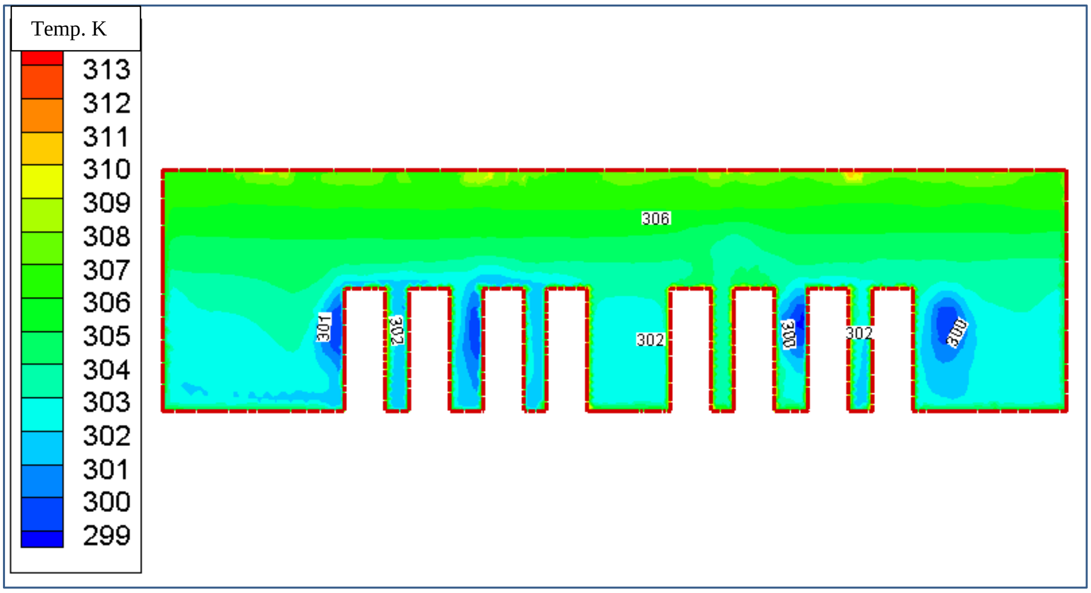
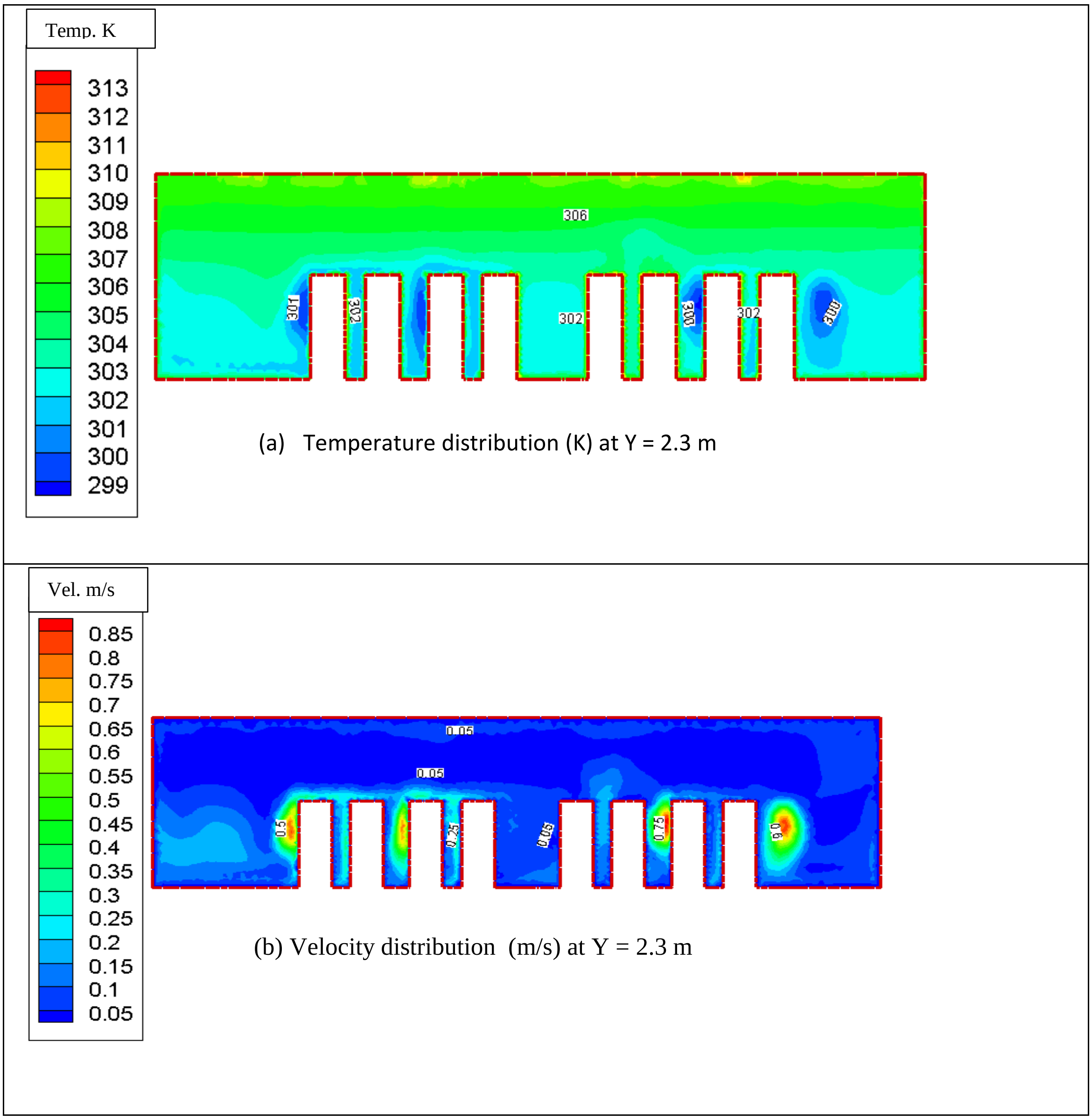


Figure 2.6 Temperature distribution (K) in environmental chamber at Y=2.3 m (Case1: Supply air temperature = 21 ℃ (or 294 K))

The distance between 4 numbers of supply & 4 numbers of return terminal in this model is dependent on the seating layout. The local room temperature at breathing zone of front row is around 28℃. The differential temperature of 1 K, in between 301 (or 28℃) to 302 K (or 28℃), as shown in Figure 2.6 is considered as even distributed profile in the breathing zoning level. It is not significant to affect the human thermal comfort result.

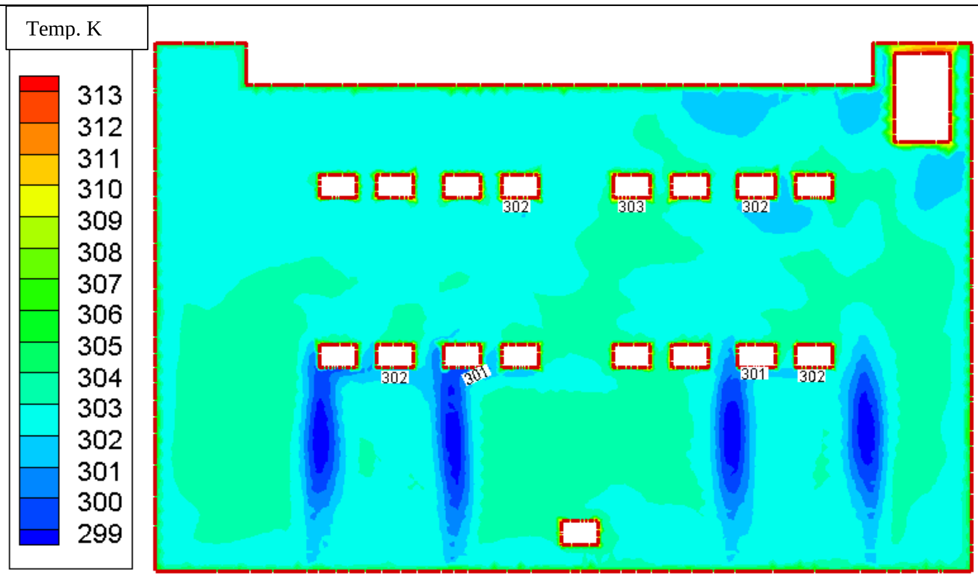
Figures 2.7 show the velocity and temperature in three different planes: Y =2.3 m and Z = 1.1 m respectively. The air velocity around the occupant is quite low, except for the region where the supply air collides with the occupant, in which the velocity is still less than 0.35 m/s. This largely prevents draught risk. Temperature stratification is clearly evidenced, with the region around the occupant being cooler than the other regions. However, the temperature gradient between floor and 1.1 m height is around 3 K.



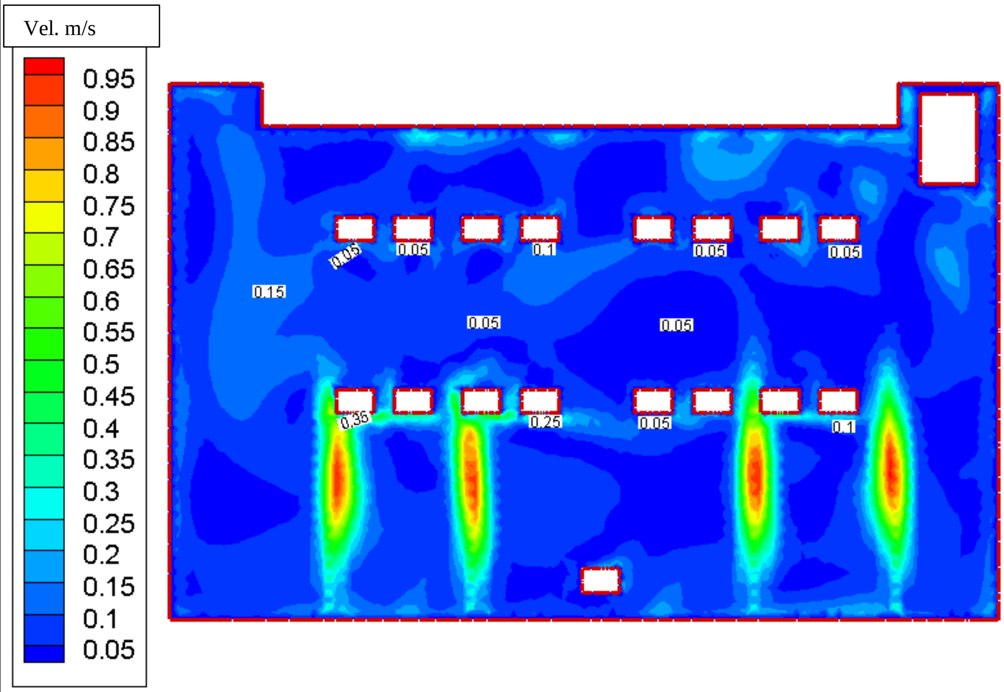
(a) Temperature distribution (K) at Y = 2.3 m

(b) Velocity distribution (m/s) at Y = 2.3 m

Figure 2.7a Temperature and air velocity distribution at Y = 2.3 m (Case 1)



(a) Temperature distribution layout (K) at Z = 1.1 m



(b) Air velocity distribution layout (m/s) at Z = 1.1 m

Figure 2.7b Temperature and air velocity distribution layout at Z = 1.1 m (Case 1)

The computed results for other three cases has similar situation with the above case 1. The detailed temperature distribution and air pattern in breathing zone at 1.1m above the floor level is shown in the case 2, case 3 and case 4 as shown on Figure 2.8, 2.9 and 2.10 respectively.

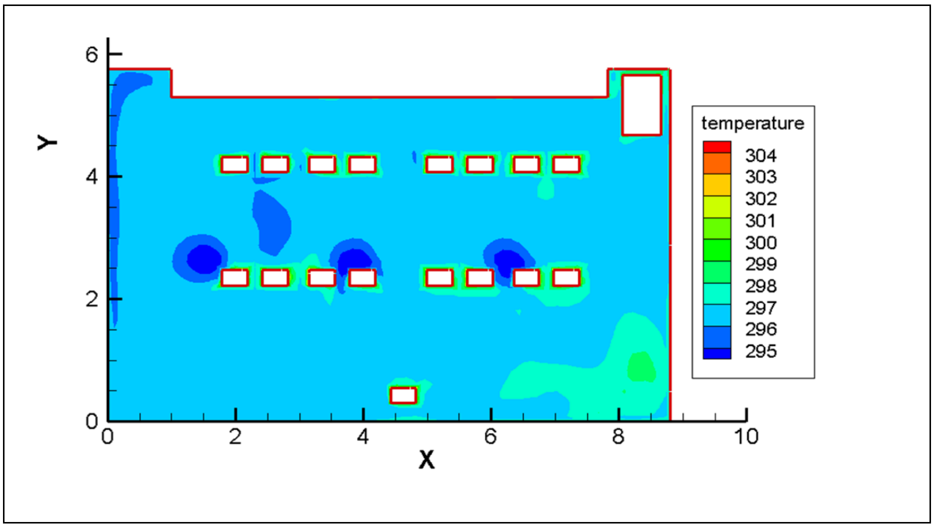


Figure 2.8a Temperature distribution layout at 1.1 m AFFL
(Case 2: MV, Supply temperature: 19.5℃(or 292.5 K); Supply velocity: 0.63 m/s)

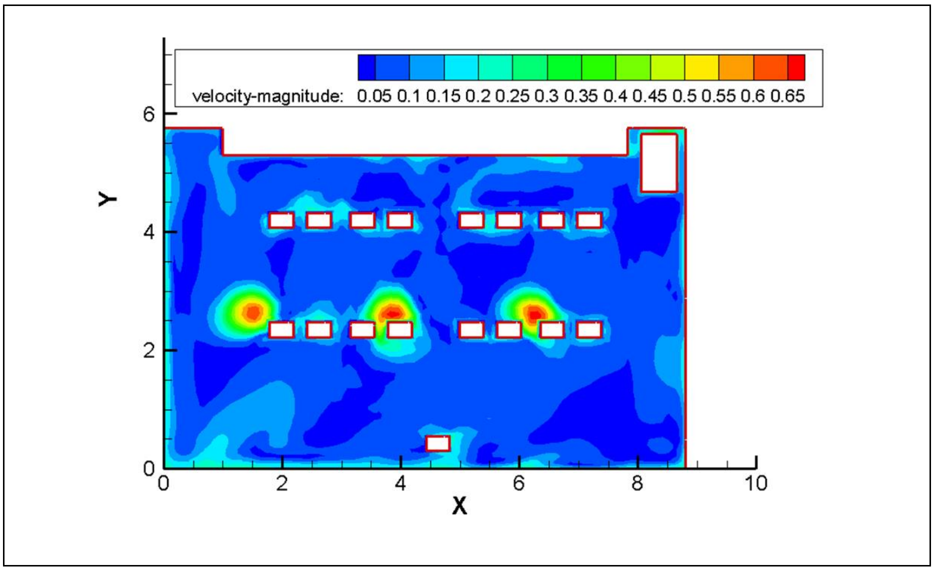


Figure 2.8b Air velocity distribution layout at 1.1 m AFFL
(Case 2: MV, Supply temperature: 19.5℃(or 292.5 K); Supply velocity: 0.63 m/s)

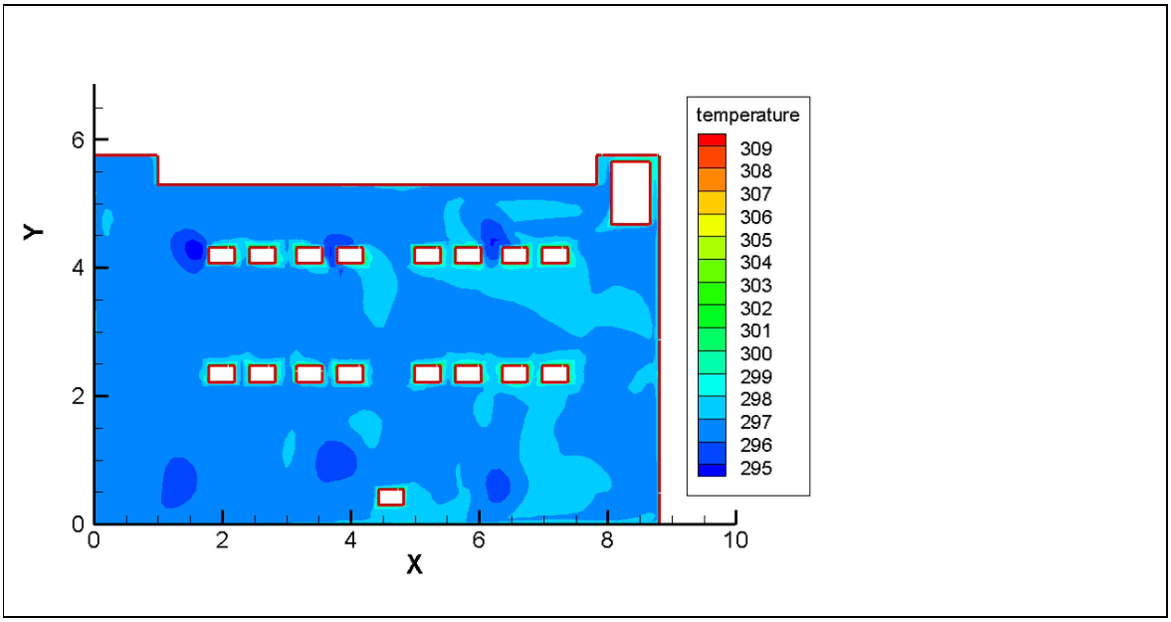


Figure 2.9a Temperature distribution layout at Z= 1.1m AFFL
(Case 3: MV, Supply temperature: 19.5℃ (or 292.5 K); Supply velocity: 0.32 m/s)

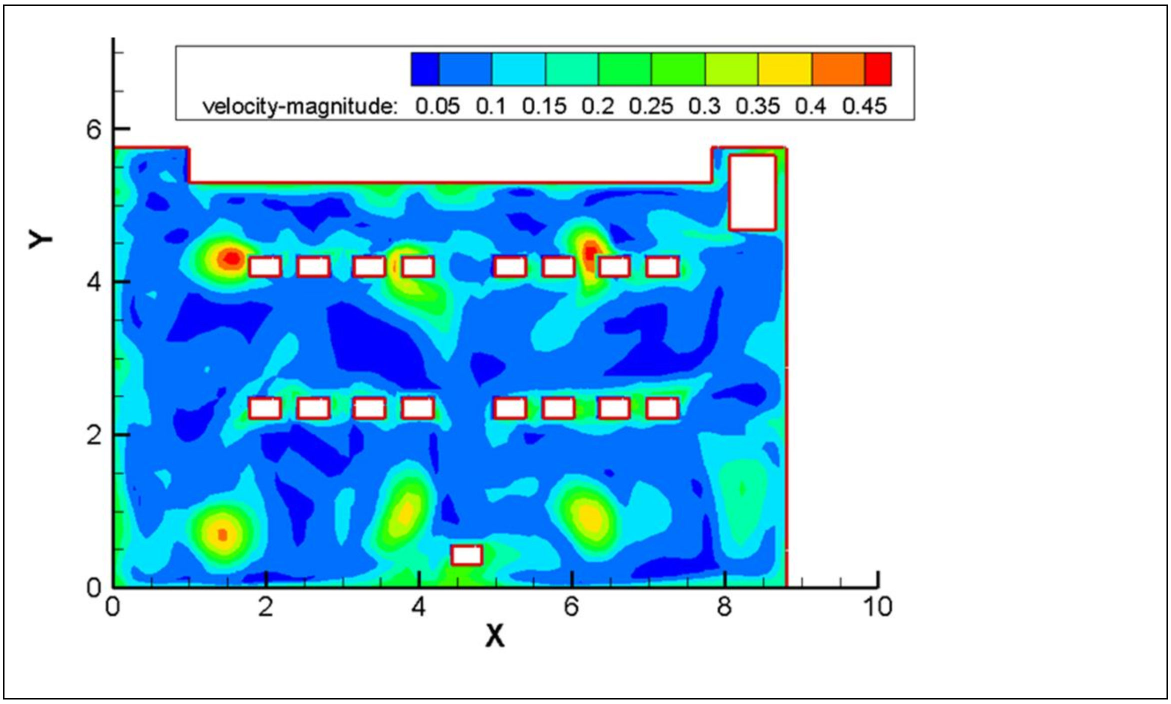


Figure 2.9b Air Velocity distribution layout at Z=1.1m AFFL
(Case 3: MV, Supply temperature: 19.5℃ (or 292.5 K); Supply velocity: 0.32 m/s)

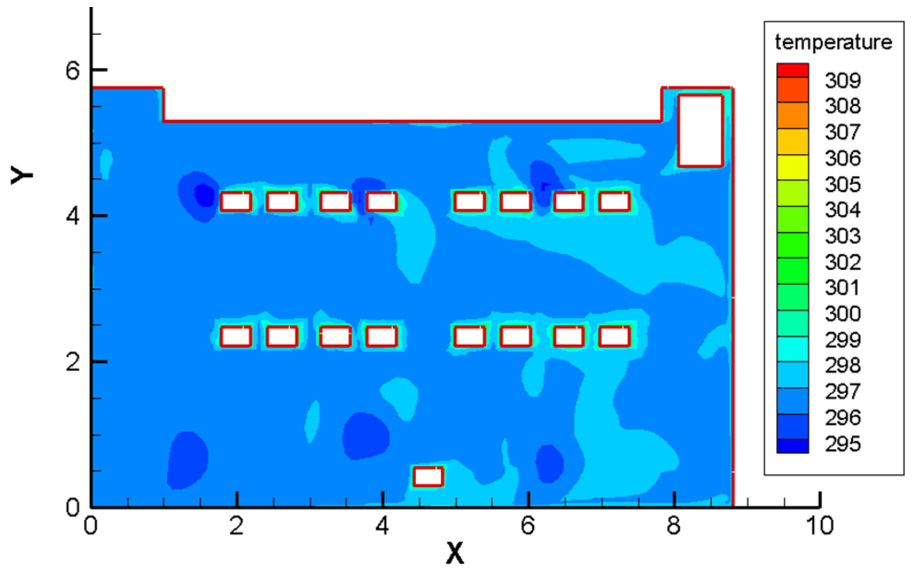


Figure 2.10a Temperature distribution layout at Z=1.1m AFFL

(Case 4: SV, Supply temperature: 19.5℃ (or 292.5 K); Supply velocity: 2.48 m/s)

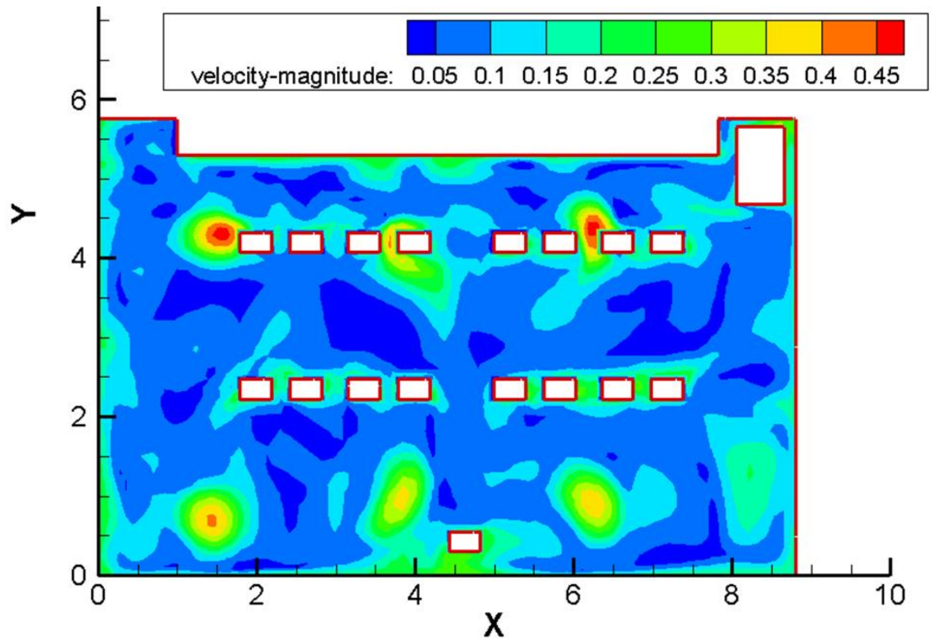


Figure 2.10b Air velocity layout at Z =1.1m AFFL

(Case 4: SV, Supply temperature: 19.5℃ (or 292.5 K) and Supply velocity: 2.48 m/s)

In the present simulation, the air does not collide with occupant exactly at the height of 1.1 m because of buoyancy. Different ventilation mode create different airflow pattern and temperature distribution layout as shown in Figure 2.7 to 2.10. In comparing with mixing ventilation, air velocity of stratum ventilation in breathing zone is significantly higher, which is about 0.8 m/s at the front row subject as shown in Figure 2.7. That is because supply air inlets of stratum ventilation are at this level. For mixing ventilation, air velocity just under the ceiling inlets is higher because of the downward supply air inlets, whereas air velocity at other locations is fairly low. This suggests that it might be improved with lower Archimedes Number, i.e., the supply air velocity is higher yet not so high as to cause draught. Therefore, while providing good air quality, thermal comfort under stratum ventilation could be improved primarily by increasing the air supply volume to lower the mean room temperature. Increasing the supply air velocity and increasing the oblique angle to the ceiling of the supply outlet are also feasible measures. The reason why the supply air velocity is low in the present simulation is to avoid the draught effect (ASHRAE55-2013). Obviously, there is a balance point for indoor air quality and thermal comfort. This model is developed in order to suit the actual local condition which is in a subtropical region. Lin et al. (2010) validated the RNG $k-\varepsilon$ model for air distributions in different buildings with displacement ventilation. The RNG $k-\varepsilon$ model has also been proved could applied for stratum ventilation. Apart from the above simulation results, the validation of uniformity tests has been studied and mentioned in Section 4.

2.1.5.1 Size of stratum ventilated room

Figure 2.11 shows a sketch of the main parameters to limit properties of jets and plumes. The related equations related to the trajectory of the flow discharged into the room and the entrainments of the flow are also described.

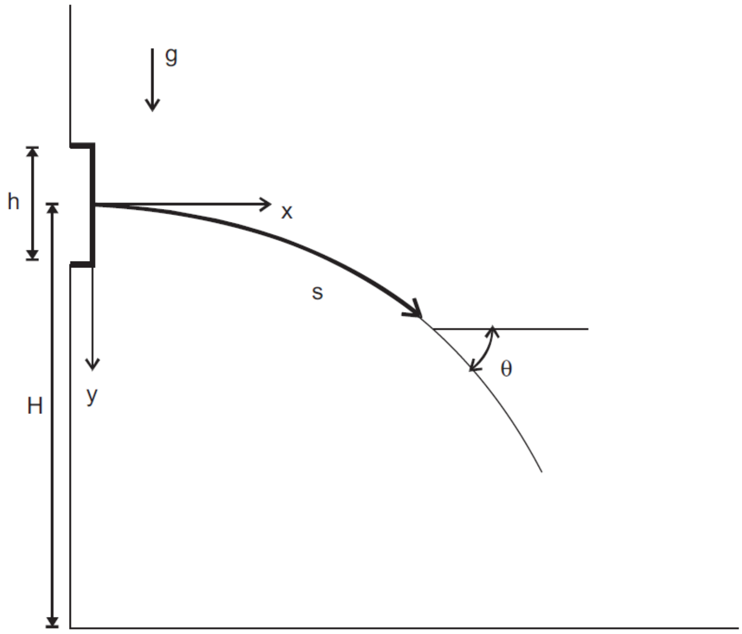


Figure 2.11 The main parameters of a buoyant jet discharged horizontally

The specific momentum flux:

$$m(0) = q(0)\,U(0), \qquad \text{Eq. (2.3)}$$

and the specific buoyancy flux:

$$B(0) = q(0)g\frac{\Delta T(0)}{T}, \qquad \text{Eq. (2.4)}$$

The horizontal component m_x of the specific momentum flux is conserved,

$$m_x = m(0), \qquad \text{Eq. (2.5)}$$

The vertical component is changed at a rate equal to the specific buoyancy flux B,

$$\frac{dm_y}{dt} = \mathrm{B}(0), \qquad \text{Eq. (2.6)}$$

After integration of the above equation with respect to the vertical component of the momentum flux after time t, by assuming that the density / temperature is uniform, no ambient cross flow and no vertical temperature gradient.

$$m_y = B(0)t, \qquad \text{Eq. (2.7)}$$

The volumetric flow rate q increase by entrainment of ambient air is

$$\frac{dq}{ds} = \mathrm{E}, \qquad \text{Eq. (2.8)}$$

The differential equation for the trajectory can be written as

$$\frac{d^2\mathrm{y}}{dx^2} = \mathrm{constant}\frac{1}{l_T^2}\frac{q(s)}{q(0)}, \qquad \text{Eq. (2.9)}$$

where l_T is the thermal length for the buoyancy jet.

Since the thermal length is measured in meters, the effect of buoyancy can be compared to the dimensions of the room and diffuser. The thermal length for a round buoyancy jet is equal to

$$l_T = \frac{U(0)A\ (0)^{1/4}}{\sqrt{g\Delta T/T}}, \qquad \text{Eq. (2.10)}$$

and for the plane boundary jet is equal to

$$l_T = \left(\frac{u(0)^4 h}{(g\Delta T/T)^2}\right)^{1/3}, \qquad \text{Eq. (2.11)}$$

For a round and plane jet with a trajectory of small curvature, the volumetric flow rate is

$$y \sim x^3, \qquad \text{Eq. (2.12)}$$

$$y \sim x^{5/2}, \qquad \text{Eq. (2.13)}$$

In practical consideration, Elvsén, et al. (2009) was conducted the test at 21℃ of mean air temperature with the supplied at a flow rate between 0.013 and 0.035 m^3/s and a temperature of 17℃ in a displacement ventilated chamber with 3.14 (l) x 4.18 (w)x 2.75 (h) m. From the above equation 2.3 to 2.13, it found that longer thermal length could be achieved by increasing air velocity, but Archimedes number is indirectly proportion to this trend. The Archimedes number was calculated according to

$$A_r(0) = \mathrm{g}\frac{\Delta T\sqrt{A(0)}}{TU_{in}^2}, \qquad \text{Eq. (2.14)}$$

and the Reynolds number was calculated as

$$\mathrm{Re} = \frac{UL}{v}, \qquad \text{Eq. (2.15)}$$

The relationship of length of the trajectory when it reaches the floor is obtained by Sanberg et al. (2009).

$$L_{Traj} = \int_0^{x(y=H)} \sqrt{\{1 + \left(\frac{dy}{dx}\right)^2} dx\}, \qquad \text{Eq. (2.16)}$$

In comparing the relationship of non-dimensional lengths L_{Traj}/L and L_{Traj}/H, the ratio between the length of the trajectory and the characteristic length of the supply device is less than 6. For an isothermal jet with a high Reynolds number a distance of about six times the characteristic demission of the supply device is required to achieve an established flow with respect to the means velocity field.

For stratum ventilation, the correlation of the room temperature, T_r to the supply, T_s and exhaust air temperatures, T_e are represented by a dimensionless temperature defined as:

$$\theta_{r=\frac{T_r - T_s}{T_e - T_s}}, \qquad \text{Eq. (2.17)}$$

According to the experimental results from Tian (2009), a value of $\theta_r = 0.726$ was derived under a supply and exhaust air temperature difference of 6.8 ℃. The product of the dimensionless temperature multiplied by the supply and exhaust air temperature difference is expected to remain fairly constant (i.e. 4.9368) for a non-industrial air-conditioned room under slightly deviated temperature differences.

$$Q_{sensible} = \mathrm{mc}(T_e - T_s)\ , \qquad \text{Eq. (2.18)}$$

It is applicable for stratum ventilation. The same form of mathematical expression was also used to present experimental results both for temperature gradient and for humidity ratio gradient by Kosonen, R. (2010). According to the above consideration, temperature difference between supply and return air temperature is major parameter to affect the thermal length, l_T. This relationship is also affect thermal comfort level to be expected by the occupancy, e.g. the supply air temperature cannot be too cool to maximize the thermal length. The size of stratum ventilated room is depended on thermal length and the main properties of buoyant, as shown in the above equation 2.10 & 2.11. Therefore, thermal comfort analysis should be investigated, and then determined its neutral temperature with an optimized supply air temperature. It is an indispensable condition to govern an actual room size serving by stratum ventilation.

2.2 Realization of environmental chamber

There were several combinations of input parameter of air distribution strategies (see Figure 2.1) together with all necessary power and control devices to be provided in the environmental chamber for conducting various human thermal tests and experimental measurement for this research. The location of environmental chamber is a core of building in order to avoid any possible external and psychological effect during this research.

2.2.1 Design features of environmental chamber

Design features of the environmental chamber include but not limited to the following:

1. Dimensions: 8.8 m (length) × 5.1 m (width) × 2.4 m (height).
2. VAV mixing, displacement and stratum systems. The air terminals at ceiling, middle and floor levels can be used either for supply or for return. Various combinations are available for experiments of different air distribution methods**.** The system consists of
   i. ceiling mounted air handling unit (AHU) with variable speed of supply fan;
   ii. carbon dioxide,CO_2 sensor for fresh air control;
   iii. air distribution device such as air ductwork, insulation, variable air volume (VAV) boxes and air diffusers;
   iv. Acoustics treatment;
   v. water distribution device such as chilled water pipe, insulation, control valves, etc.;

vi. monitoring and control device for assessing thermal comfort and energy consumption;
vii. control and monitoring devices such as temperature sensors, humidity sensors, and energy meters.
viii. an enthalpy control sensor to automatically reset the indoor design conditions;
ix. interface to the measurement parameters with the building management system (BMS).

3. Central chiller plant operations simulator with various system circuits, including differential pressure bypass, primary-secondary and variable primary flow. The simulator should be able to simulate
   i. the operation sequence of various types of chiller combination, such as single chiller and/or multiple chillers and the associated circulation pumps; and
   ii. full load and part load control philosophy for various chiller combinations.
4. Building management system
   i. Real time power information: rms values of phase & line voltages and currents, displacement and total power factors and the leakage currents.
   ii. Energy data: kWh, maximum kW, kVA and kVAr with time stamps
   iii. Event data: minimum/maximum values with time stamp up to 1 second record for the above items (i) and (ii), sag/swell voltage or set point current trigger up to half cycle calculation and transient events. On/off/trip and busbar temperature monitoring are also included in the critical event monitor points as well.

iv. Power harmonics: THD of voltage and current harmonics; individual odd current harmonic measurement may extend to the 21st and neutral to earth voltage.

v. workstations with web server, historian server, IE Browser, wireless and wired access control and monitoring devices and LED display panel with the size of 42 inches. The wireless applications can be effected by UMPC and PDA.

vi. standardized control and communication means for various systems, using BACnet protocol, MODBUS and/or RS232.

Each present parameter can be identified and controlled via the remote link of "http://144.214.83.223//DeltaWeb/login.asp" of the building management system which is shown in Figure 2.12.

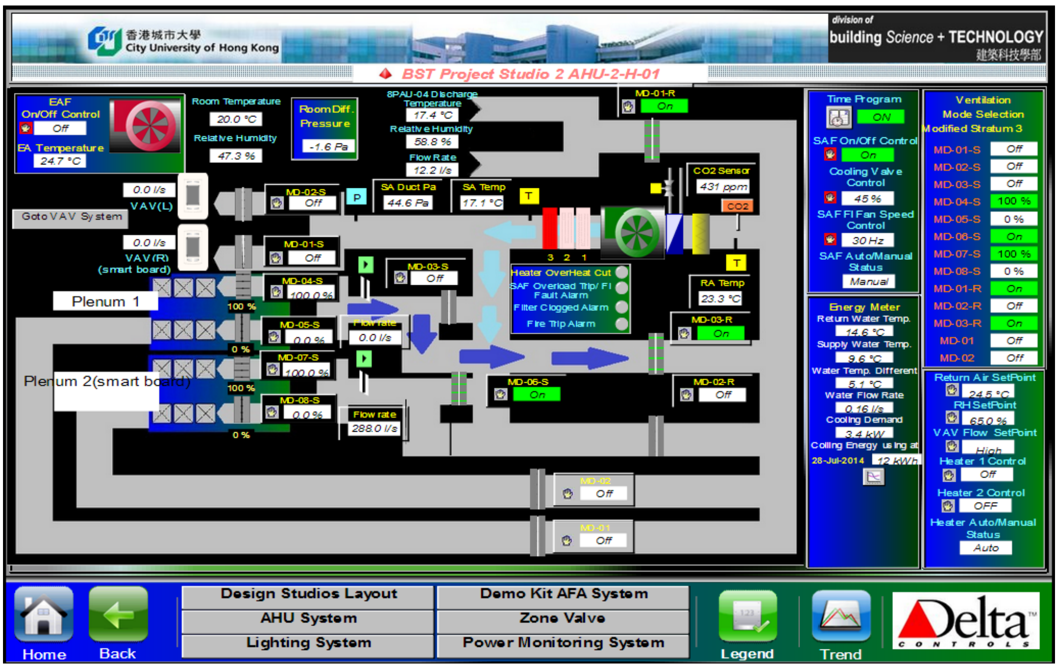


Figure 2.12 BMS display for the combinations of air distribution strategies

The above mentioned features have been constructed by term contractors under my supervision via Campus Development Facility Office; as well as all necessary equipment/devices, such as BMS & smart meter supplier with totally approved cost of HK$ 1,850,000 which were included:

a) HK$ 697,450 for building services installation by internal term contractor;
b) HK$ 702,425 for this building management system by BMS supplier - Hensen System Engineering Limited;
c) HK$ 188,676 for Power Quality and Energy Monitoring System by smart meter supplier - Powerpeg NSI limited;
d) HK$ 135,000 for BACnet integration for exiting BMS by Johnson Controls;
e) HK$ 31,890 for lighting system by GELEC(HK) Limited; and
f) HK$ 94,559 as provisional sum for history server, web server, operation workstation and wireless router, UMPC & PDA devices for Building Management System.

This environmental chamber has been ready for conducting the research work since 2010. The detail of my construction work together with testing and commissioning process of this environmental chamber is shown in Appendix A.

2.3 Experimental setup

My study used the layout of an environmental chamber 8.8m x 6.1m x 2.4m (length x width x height) in Figure 2.13, larger than that at Xi'an Jiaotong University (Tian et al. 2011) which was used in the preliminary study. This will form the basis to develop an in-scale model of the environmental chamber for the study of thermal comfort and cost effectiveness of six air distribution strategies.

For mixing ventilation, supply air is circulated to the occupied zone from six ceiling supply diffusers and three return air louvers at 2.4 m above the floor level. For displacement ventilation, supply air is provided from both sides of 4 wall mounted diffusers at 0.3 m above floor level and returns to 3 ceiling supply terminals. For stratum ventilation modes, air is supplied horizontally from four perforated wall-mounted type diffusers installed in the front wall at 0.3 m or 1.3 m above the floor level together with 4 wall mounted return air terminals in the rear or front wall at 1.3 m or 0.3 m above the floor level. The different levelling of terminals at 0.3 m and 1.3 m has been selected to test which air distribution strategies is better for stratum ventilation to elevated the indoor temperature without sacrifice of thermal comfort level. Each supply and return air terminal has been well commissioned to ensure a uniformity of air distribution under six ventilation modes.

2.3.1Control and measured parameters

According to ASHRAE 55-2010, there are six primary factors to define a thermal comfort conditions, including (i) metabolic rate, (ii) clothing insulation, (iii) humidity, (iv) radiant temperature, (v) air temperature, and (vi) air velocity. The first three factors are controlled during thermal comfort survey. Since the environmental chamber was surrounded by air-conditioned spaces, the walls, the floor and the ceiling are assumed to be adiabatic. A Thermal infra-red tracer (NEC TH7800N) was used to monitor the room surface temperature against the desired room temperatures for evaluating the effect of the radiant temperature on thermal comfort. My experimental study can only focus on the combination of last two factors to estimate the neutral temperature during thermal comfort survey.

During human comfort tests, three parameters including air temperature, air velocity and humidity level are continuously controlled and measured, by means of INNOVA data logger at occupancy breathing zone and existing air conditioning system via building management system, in order to ensure the room condition under steady state. Five measurement points are illustrated in Figure 2.13.

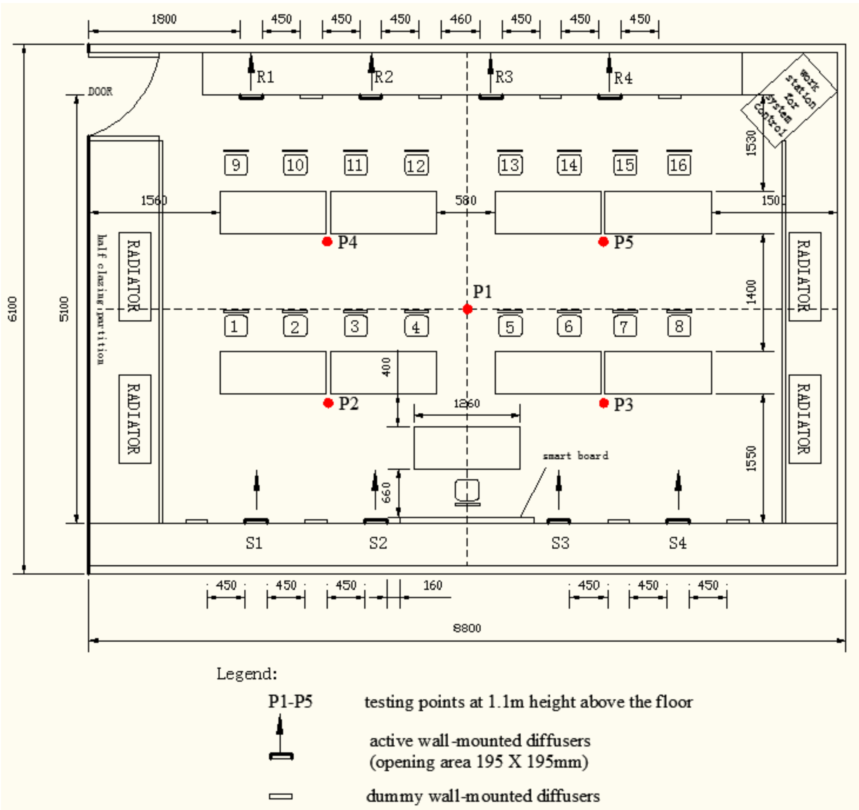


Figure 2.13 Layout of environmental chamber

Due to the surrounding area of this chamber was fully air-conditioned, a certain numbers of radiators were used to overcome any undesirable heat transfer through the partition wall. For example, radiators were placed near the left and right walls in the room in order to maintain 28℃ room temperature for certain sessions. The temperature used in thermal comfort experiments, which is correlated with the *PMV* scale (to find neutral temperature), is the mean air temperature in the occupied zone.

In conducting human comfort tests, all temperatures mentioned in Tables 2.4 are measured and monitored by the central monitoring sensors positioned 0.1 m beneath the geometrical center of the environmental chamber at 1.1 m above the floor (Figure 2.14). Duct-type air temperature sensors installed inside the main supply and return ducts are logged through the standalone building management system in order to ensure the chamber is in steady state. The air temperature at the 1.1 m level is correlated to the *PMV* scale to find out the neutral temperature for the three ventilation methods.

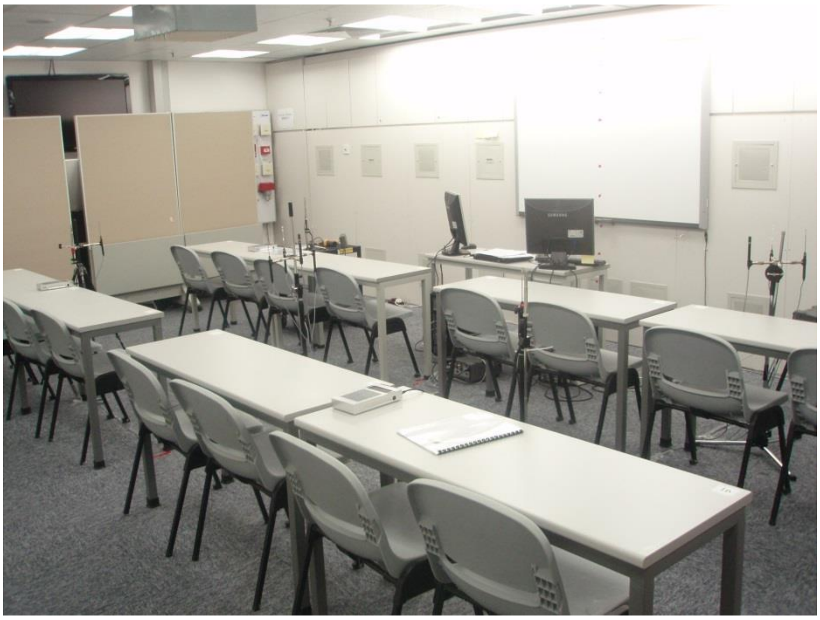


Figure 2.14 Sensors location at 1.1 m AFFL

Each test session involves sixteen subjects, who sit by a working desk in four arrays. Each test session is designed to acquire the subject responses at specific room temperature, supply airflow rate and ventilation method. Then the next session is repeated with a different subject group. The room parameter setting is altered in the sequence of 28℃→ 26℃→ 24℃, and 10 ACH → 15 ACH.

To ensure that the room condition is steady for every session, additional sensing devices with a data logger are employed to monitor all relevant parameters such as air temperature, air speed, operative temperature, effective temperature, relative humidity at 1.1 m above the floor level. Those parameters shall be steady throughout the test sessions in order to minimize any uncertain factor. The actual room conditions are determined using an INNOVA professional measuring tool set with data logger. LUMASENSE transducer MM0038 is used for measuring air velocity. The measuring range is 0 to 10 m/s; the accuracy is ± ($0.05v_a$ + 0.05) m/s; the dynamic responding time is 0.2 s. LUMASENSE transducer MM0034 is used to measure air temperature with an accuracy of ± 0.2℃. LUMASENSE transducer MM0060 is used to measure operative temperature. The accuracy is ± 0.3°C. LUMASENSE transducer MM0037 is used to measure humidity based on dew-point temperature with an accuracy of 0.5 %RH. The details of anthropometric data of test subjects, as well as the control & measured parameters of each experimental session will be addressed in the later Chapter 3. For the highest air speed of 0.76 m/s, the room temperature has to be 25.3℃ or higher to avoid draft risk according to Addenda *d, e, f,* and *g* to ANSI/ASHRAE Standard 55-2010.

2.3.2 INNOVA data Logger

Human comfort tests of thermal sensations with INNOVA data logging station were conducted under mixing ventilation, displacement ventilation, and stratum ventilation. The module 1221 of INNOVA data logger was used to collect data as a stand-alone apparatus located within occupancy zone. It was connected with a personal computer and 7301 software via its serial interface to view data on-line. This module enables the connected transducers to provide measurement data for the majority of the physical parameters used in evaluating thermal comfort such as (in accordance with ISO7730) *PMV*, *PPD* and *DR*.

The module has three sockets for the following transducer types:

- o Temperature (MM0034, MM0035, MM0060)
- o Humidity (MM0037)
- o Air Velocity (MM 0038)

The input sockets for temperature and humidity transducers have a measurement range from –20 to 100℃, with a resolution of 0.1℃. The MM 0037 is a very fast responding sensor. During the active part of the measuring cycle, it responds within 1 to 2 seconds, as opposed to more conventional humidity transducers which typically have response times of many minutes. This very fast reaction time can often come as a surprise if you are not aware of the fact that humidity may be very varying both in time and space. This is particularly true close to the body and breath. This equipment is shown on Figures 2.15 to 2.18.

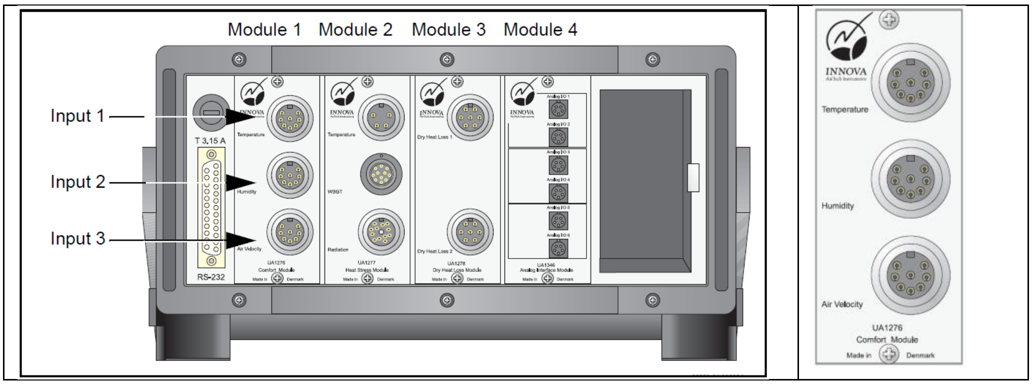


Figure 2.15 Module/transducer input position numbers on 1221& Comfort Module UA1276

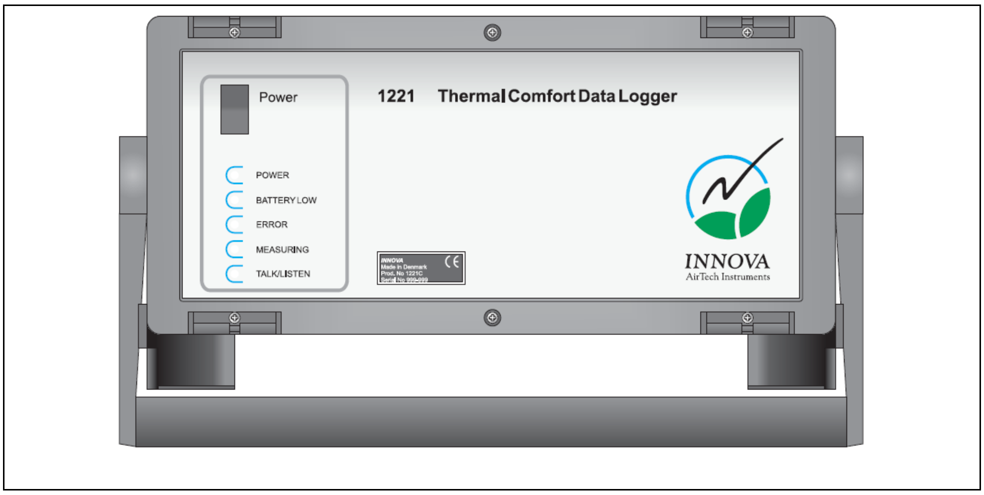


Figure 2.16 Thermal comfort data logger

Figure 2.17 Outlook of module/transducer

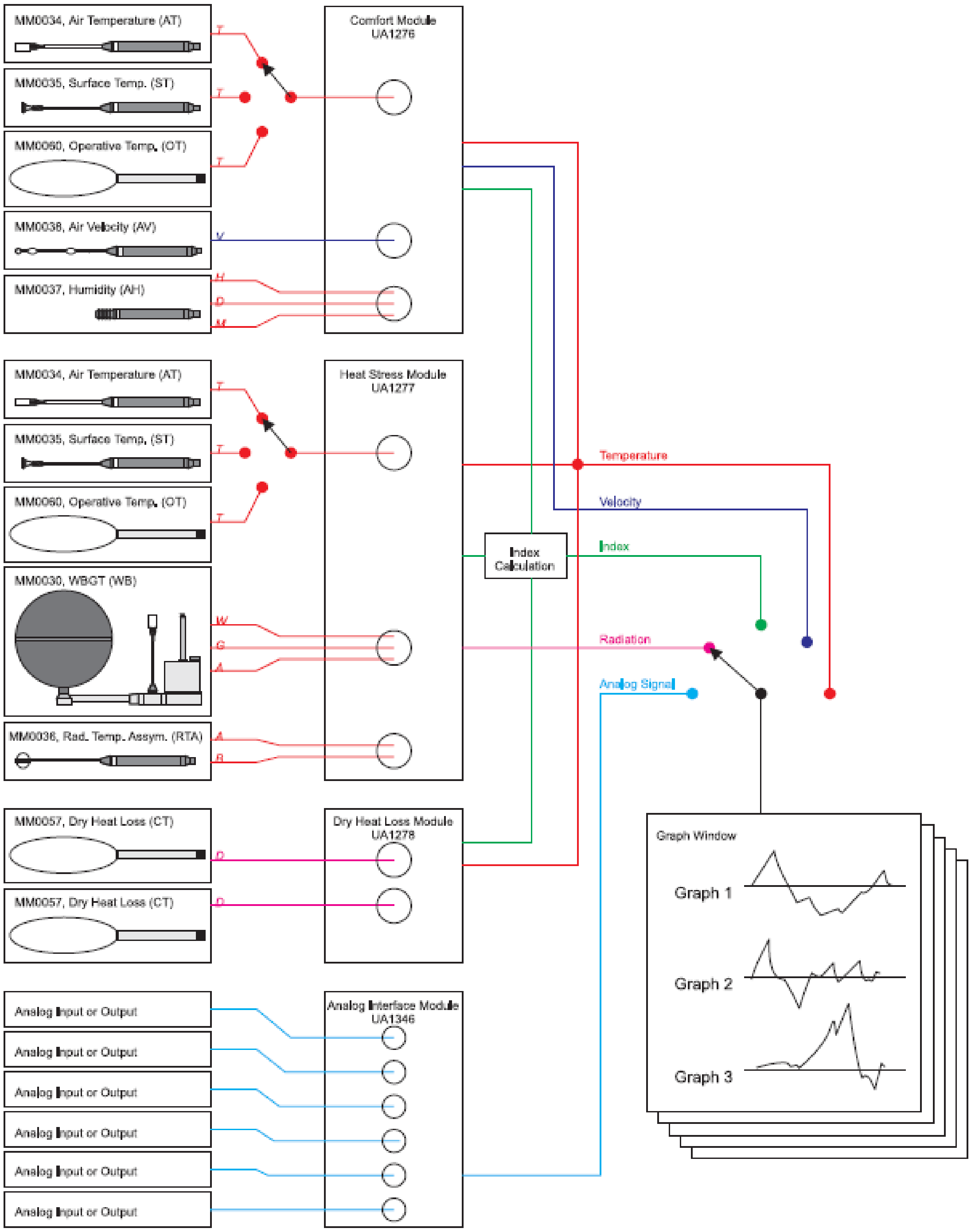


Figure 2.18 Overview of transducer types used with each 1221 module

2.4 Research and teaching platform

This environmental chamber was installed into the existing Design Studio of the Division of Building Science & Technology (BST), City University of Hong Kong. This installation was consists of a building management system (BMS) as shown in Figure 2.19 with open BACnet protocol, a power quality monitoring system, a addressable lighting system, an automatic fire alarm system, an mixing/displacement/stratum ventilation systems. By means of these facilities, a researcher would be able to familiarize with the operation of these intelligent building systems, to have hands-on experiences on collecting all necessary data for further investigation, as well as to appreciate various energy saving initiatives.

The installation work is also an excellent teaching and learning tool to provide researchers and students with broad-based academic foundation and practical works in building services engineering to enter into an international workplace or continuing education and research in local and overseas universities. The animation of mixing, displacement and stratum serving the classroom are uploaded into my local university's website. One of evidence is my student won the first prize of Samsung Smart Education competition in May 2014. The detail is shown in Appendix B1.2.

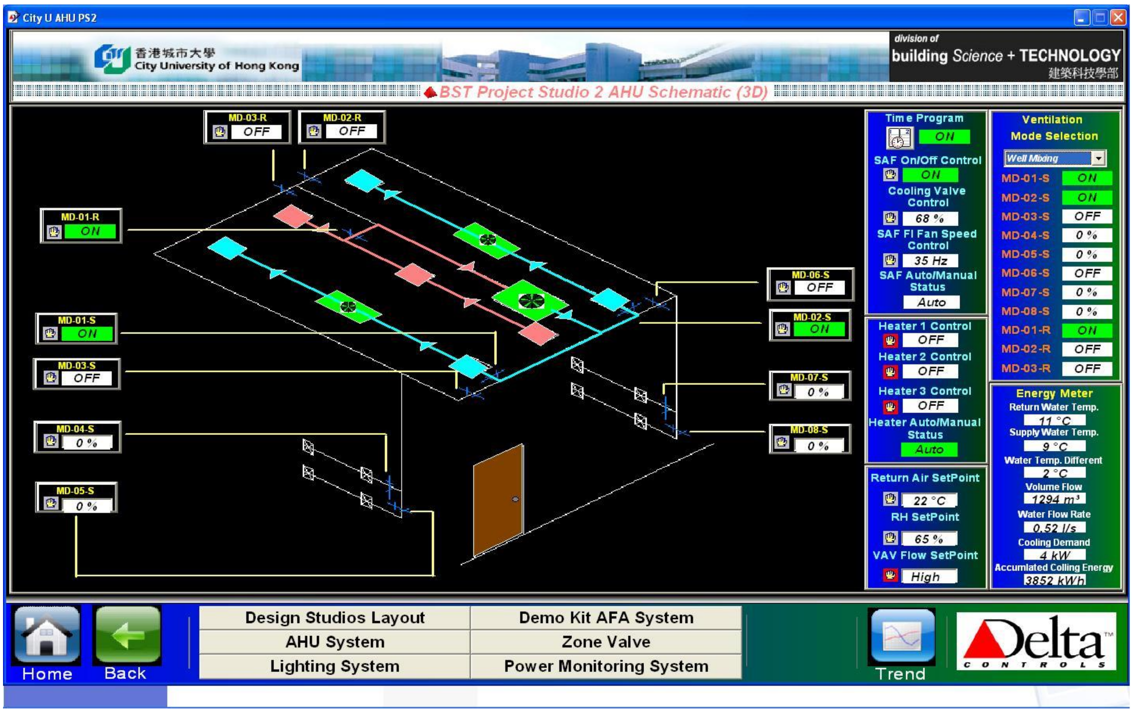


Figure 2.19 Three Dimensions of ventilation systems inside environmental chamber

2.5 Concluding remarks

This environmental chamber of 8.8 m (length) × 6.1 m (width) × 2.4 m (height) with all data collection equipment/device has been setup for conducting human comfort test and experimental field measured data collection. Computational fluid dynamic simulation was used to assist and finalise the necessary information for built-up this environmental chamber, such as the location of supply & return terminals, air parameters, heat source identification and furniture arrangement, etc.

The features of the environmental chamber included mixing, displacement and stratum ventilation modes consisting of, but not limit to, the air terminals at ceiling, middle and floor levels can be used either for supply or for return; ceiling mounted AHU with variable speed of supply fan; CO_2 sensor for fresh air control; air distribution device such as air ductwork, insulation, VAV boxes and air diffusers; acoustics treatment; water distribution device such as chilled water pipe, insulation, control valves; monitoring and control device for assessing thermal comfort and energy consumption; control and monitoring devices such as temperature sensors, humidity sensors, and energy meters; an enthalpy control sensor to automatically reset the indoor design conditions; interface to the measurement parameters with the Building Management System (BMS). The building management system consists of real time power information in term of r.m.s values of phase & line voltages and currents, displacement and total power factors and the leakage currents; energy data in terms of kWh, maximum kW, kVA and kVAr with time stamps; workstations with web server, historian server, IE Browser, wireless and wired access control and monitoring devices and LED display panel with the size of 42 inches. The wireless applications can be effected by smart phone and any electronic devices of which standardized control and communication means for various systems, using BACnet protocol, MODBUS and/or RS232. Apart from the BMS, five measurement points were also setup within the breathing zone during human comfort test, three parameters including air temperature, air velocity and humidity level were continuously control and measured by the INNOVA data logger and building management system in order to ensure the room condition under steady condition.

The detailed analysis of all collected experimental measurement and validation of the numerical result, as well as, estimating the energy consumption of six ventilation modes are described in the following chapters.

CHAPTER 3: THERMAL COMFORT DATA ANALYSIS

3.1 Method for conducting the thermal comfort survey

Human comfort tests were conducted to evaluate the neutral temperature of the thermal environments in a mixing, displacement, stratum ventilated chamber with dimensions of 8.8 m (L) × 5.1 m (W) × 2.4 m (H). For mixing ventilation, the supply and return terminals are installed at the ceiling level (Figure 1.1); for displacement ventilation, the supply diffusers are located at the front and rear wall plenum and the return at the ceiling level (Figure 1.2); and for stratum ventilation, supply is made to diffuse from front sides of wall outlets at mid-level and return to low level wall inlets (Figure 1.3).

Forty-eight college-age human subjects (24 women and 24 men) are employed to participate in this research work. Most of the subjects are students of City University of Hong Kong and in good health. All of them are between 20 and 23 years old and either born or stayed in HKSAR for more than 10 years. The subjects report that they are used to staying in an air-conditioned environment. This test is conducted at the summer break, and all subjects received remuneration for their contribution. The anthropometric data of the subjects are shown in Table 3.1.

Table 3.1 Anthropometric data of subjects participating in my research work

Gender	Statistics	Age	Height , H (m)	Weight ,W (kg)	BMI* (kg/m^2)	BSA** (m^2)
Male	Mean	21.1	1.73	64.4	21.5	1.8
	S.D.	0.9	5.0	9.5	2.8	0.1
Female	Mean	21.3	1.58	50.1	20.1	1.5
	S.D.	0.9	3.5	5.2	2.0	0.1
Male + and Female	Mean	21.2	1.66	57.3	20.8	1.6
	S.D.	0.9	8.6	10.5	2.5	0.2

* Body Mass Index (BMI) = W / H^2, W is weight in kg, H is height in m.
** Body Surface Area (BSA) = $(W^{0.425} \times 100\ H^{0.725}) \times 0.007184$, W is weight in kg, H is height in m.

Before participating in this test, each subject has been briefed about the details of every step of the experiment and about the meaning of each question in the questionnaire and in addition to collect all necessary basic information.

Figure 3.1 Human comfort test session

They are briefed to wear local typical summer clothing such as short-sleeve shirts, long trousers, underwear, socks, and shoes and to carry out reading activities during this test (Figure 3.1). Therefore, the clothing insulation value and activity level are limited to 0.57 clo and 1.0 met, respectively. A thermal comfort model using the predicted mean vote (*PMV*) was derived from the steady-state heat balance of a human body with empirical constants determined from chamber studies for thermal sensation (Fanger, 1972, ISO7730) (European Standard EN ISO 7730-2005).

3.2 Data collection in different settings

A series of sessions of the entire thermal sensation test with 16 subjects takes about 3 hours and performed during 9 am to 12 noon, 1–4 pm, and 5–8 pm. Each session (a combination of settings) is designed to acquire the subject's responses at a specific airflow rate and room temperature at the range of 50%RH to 63% RH under different ventilation methods. Three temperature settings (24℃, 26℃, and 28℃) and two supply airflow rates:10 and 15 air change per hour (ACH) under mixing ventilation, displacement ventilation, and stratum ventilation are used in this study. Each of the three subject groups: (1) 16 males, (2) 16 females, (3) 8 females plus 8 males are in turn involved in a series of the 18 test sessions as shown in Table 3.2. 8 subjects are replied to vote in males and females session. Each test setting is repeated three times with different subject groups.

Table 3.2 Combination of three experimental sessions

Session (no. of participant and gender)	Combination of each parameter setting		
	Ventilation mode	ACH	Temp.
1 (16 males); 2 (16 females); 3 (8 males + 8 females)	Mixing & Displacement & Stratum x 4	10 & 15	24℃ & 26℃ & 28℃

For each session, the subjects are required to wait in another air-conditioned room near the chamber for 10 min before the scheduled time. They are told to bring their own reading material or choose any reading materials provided by the research staff during the test.

A questionnaire is adopted in the test (Figure 3.2), which concerns the thermal sensation vote of the ASHRAE 7-point scale, i.e. -3, -2, -1, 0, +1, +2, and +3 representing cold, cool, slightly cool, neutral, slightly warm, warm, and hot, respectively. During each 30-min session under steady condition, each subject is asked to mark down his/her instantaneous thermal sensation, together with their opinions on the air temperature, speed, and humidity in every 5-min interval. The six completed questionnaires are collected for each session help to assess the quality of the final response, which is subsequently used in the analysis. It was found that the last two votes were most consistent in respect with their trend of six thermal comfort voted in questionnaires. All of subjects are not know an actual setting of temperature, supply flow rate and ventilation mode in the chamber during each session.

Thermal Comfort Survey Form

Date: ____________ **Seating Number:** ________ **Form No.** ________

Part A - Feedback of Thermal Sensation

Thermal Sensation						
Cold	Cool	Slightly cool	Neutral	Slightly warm	Warm	Hot
-3	-2	-1	0	+1	+2	+3

Part B - Reason of thermal comfort/discomfort

Temperature		
Low	Neutral	High

Relative Humidity		
Low	Neutral	High

Air Velocity		
Low	Neutral	High

Part C - Acceptability

☐ Acceptable ☐ Unacceptable

Figure 3.2 Sample of questionnaire for human comfort survey

3.2.1 Monitor of room condition

Five measurement points are shown in Figure 3.3. In conducting current human comfort tests, all temperatures, supply flow rate, air speed, and relative humidity are measured and monitored by the central monitoring sensors positioned 0.1 m beneath the geometrical center of the environmental chamber at 1.1 m above floor level. The results are listed in Table 3.3a and 3.3b. Due to the surrounding area of this chamber was fully air-conditioned at 24℃, a certain numbers of radiators were used to overcome any heat loss through the partition wall in order to maintain 28℃room temperature for certain sessions. Radiators have been placed near the left and right walls in the room to provide an additional heating load in the test chamber to increase the space heat.

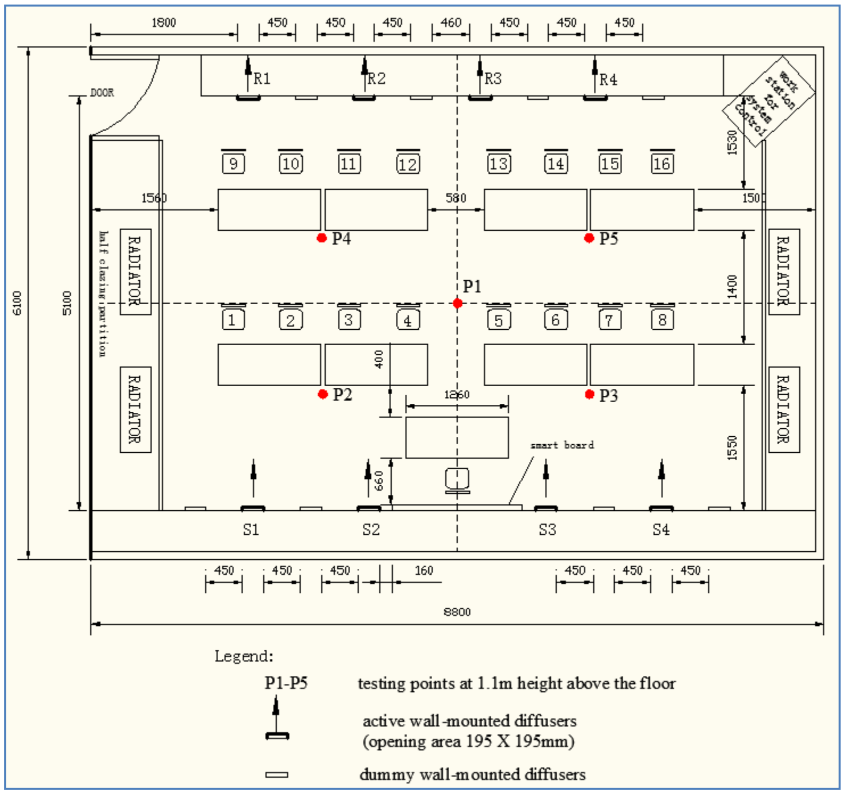


Figure 3.3 Five measurement points inside environmental chamber

Table 3.3a Control and measured parameters of three experimental sessions for six ventilation modes

Session (no. of participants and gender)	Nominal room temperature (℃)	Actual room temperature (℃)	Supply face velocity (m/s)	Air supply temperature (℃)	Measured RH (%)
1 (16 males)	24	24±0.5	1.73 [10 ACH]	15.5	51-57
2 (16 females)	26	26±0.5		16.9	52-57
3 (8 males + 8 females)	28	28±0.5		21.1	52-63
1 (16 males)	24	24±0.5	2.54 [15 ACH]	15.5	51-61
2 (16 females)	26	26±0.5		16.9	50-63
3 (8 males + 8 females)	28	28±0.5		21.1	51-58

Table 3.3b Measured air speed of three experimental sessions

Air Change per hour	Ventilation Method	Range of measured air speed (m/s) at Centre of room at 1.1m AFFL
10	Mixing	0.08-.012
	Displacement	0.02-0.05
	Stratum	0.15-0.42
15	Mixing	0.1-0.14
	Displacement	0.02-0.07
	Stratum	0.52-0.76

Duct-type air temperature sensors installed inside the main supply and return ducts are logged through the standalone building management system (Figure 3.4) in order to ensure that the chamber is in a steady state. The air temperature at the 1.1 m level is correlated to the *PMV* scale to identify the neutral temperature of the six ventilation modes.

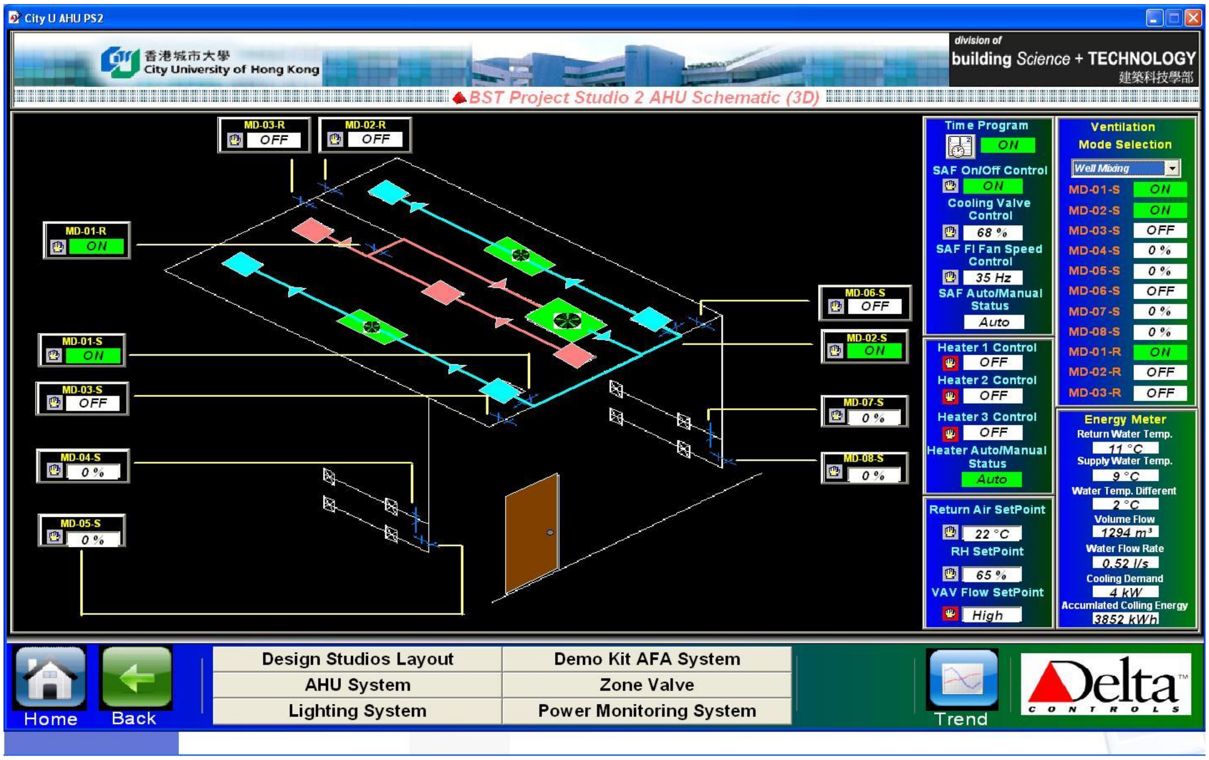


Figure 3.4 Room condition to be monitored by BMS

According to ASHRAE 55-2010, there are six primary factors to define a thermal comfort conditions, including (i) metabolic rate, (ii) clothing insulation, (iii) humidity, (iv) radiant temperature, (v) air temperature, and (vi) air velocity. The first three factors are controlled during thermal comfort survey. Since the environmental chamber has been surrounded by air-conditioned spaces, the walls, the floor and the ceiling are assumed to be adiabatic. A Thermal infra-red tracer (NEC TH7800N) has been used to monitor the room surface temperature against the desired room temperatures for evaluating the effect of the radiant temperature on thermal comfort (Figure 3.5). Thus, my experimental study can only focus on the combination of last two factors to estimate the neutral temperature during thermal comfort survey. According to the monitoring device, the surface temperature at ceiling lighting (Figure 3.6) and each air terminal at wall (Figure 3.7) were close to the required room air temperature settings as shown in above Table 3.3a. The result shows that the wall surface temperatures are close to the room temperature, indicating that the influence of the wall radiant temperature on thermal comfort is negligible. This can ensure the uniformity and steady condition of each human thermal comfort test.

Figure 3.5 Thermal infra-red tracer

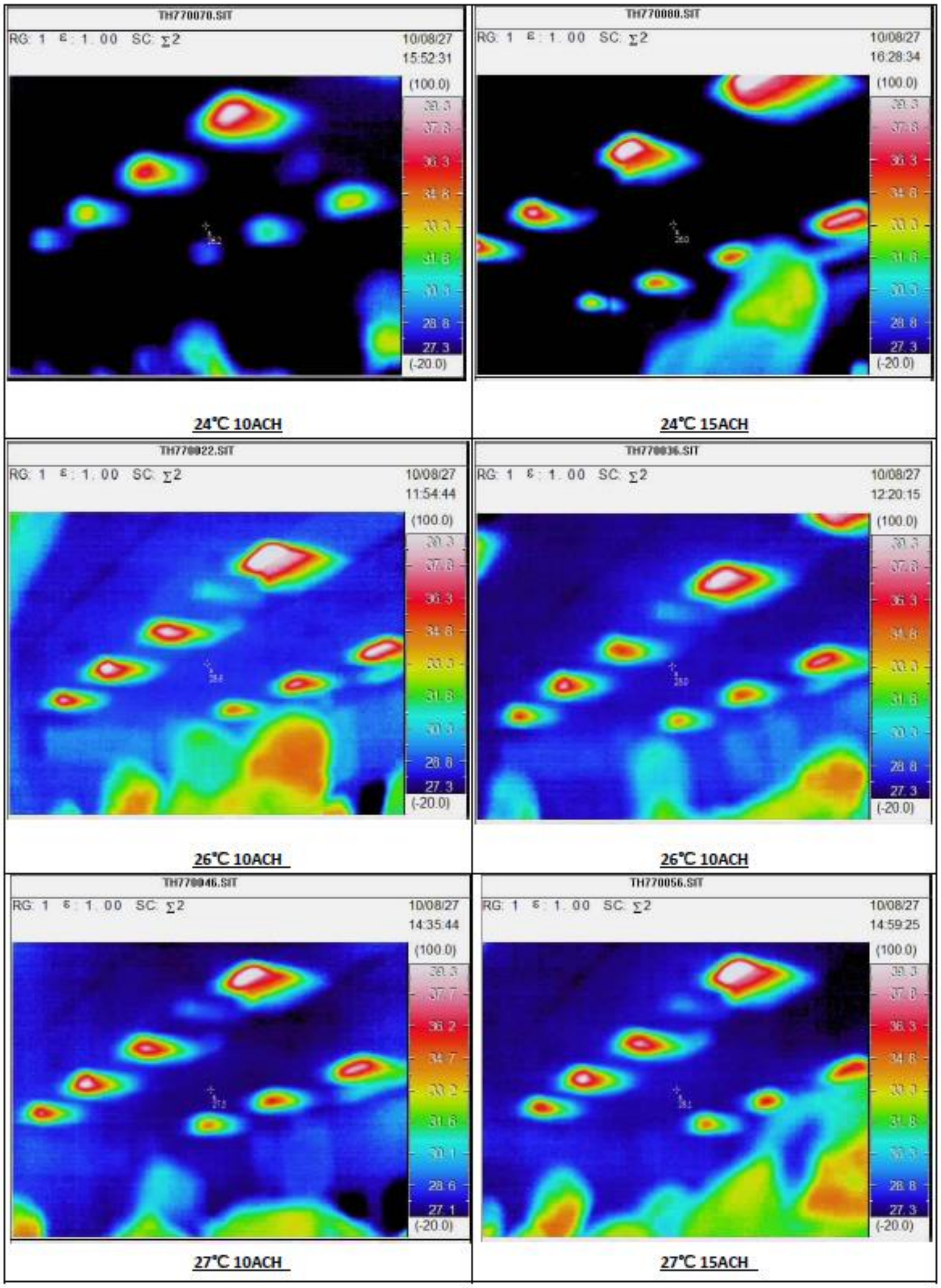


Figure 3.6 Surface temperatures at ceiling lighting

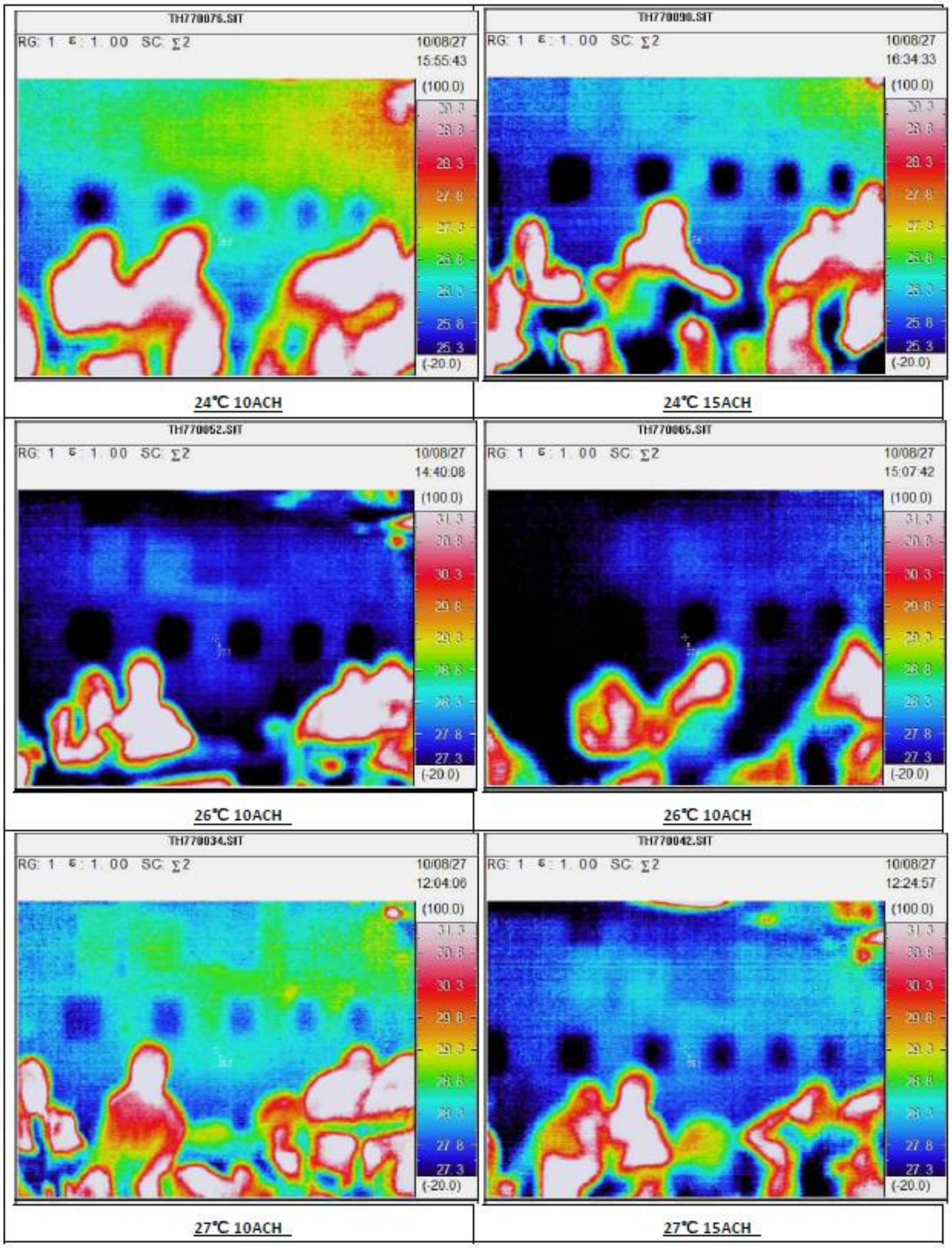


Figure 3.7 Surface temperatures at wall

3.3 Distribution of acceptance percentage

Score (-1), (0), and (+1) under a 7-point scale in the survey of thermal satisfaction conditions are defined as acceptable conditions. The acceptance percentage is calculated from the subjects selecting a score between -1 to +1. A survey of 48 subjects doing the actual sedentary activity and wearing the actual standard clothing under a range of specific temperatures 24℃, 26℃, and 28℃ was performed. Therefore, the clo-value and activity level are limited to 0.57 and 1.0 respectively. The acceptance rate of neutral (0) will be compared in the six ventilation modes under specific boundary conditions. Unacceptance or a dissatisfied percentage is defined as those whose scores are -2 (cool) or -3 (cold), + 2 (warm) or + 3 (hot). More intense discomfort is expressed by those rating higher than + 2 or lower than -2. These results are identified as dissatisfactory scores. The distribution of the rating for the thermal sensation from 48 subjects in the six ventilation modes under the specific room conditions, a combination of 24℃, 26℃, and 28℃, and 300 l/s and 450l/s of air flow supply rate, is tabulated in Table 3.4. **Bold** figure represent highest acceptance percentage, **Bold** & <u>underline</u> figure represent highest percentage voted (0) neutral and of each ventilation mode. In comparison with all ventilation modes, the highest acceptance percentage of 100% is at 26℃ & 10 ACH under stratum ventilation and 28℃ &15 ACH under modified stratum-3 ventilation. The two highest percentage voted (0) neutral are 86.2% at 26 & 10ACH under displacement ventilation, and 84.4% at 28℃ &15ACH under modified stratum-3 ventilation. The co-relationship between the acceptance percentage, temperature and flow rate for each ventilation mode is illustrated in Figures 3.8 to 3.13.

Table 3.4 Distribution of thermal sensation votes under six ventilation modes at 10ACH and 15 ACH

Ventilation Mode	ACH	Temp., °C	Average room Temp., °C	Average acceptable percentage (-1), (0), (+1)	Average unacceptable percentage (-3), (-2), (+2), (+3)	Cold (-3)	Cool (-2)	Slight Cool (-1)	Neutral (0)	Slight Warm (1)	Warm (2)	Hot (3)
Mixing (MV)	10	28	28	71.1%	28.9%	0.0%	0.0%	0.0%	22.5%	48.6%	21.7%	7.2%
		26	25.79	95.3%	4.7%	0.0%	0.0%	3.6%	67.8%	23.9%	4.7%	0.0%
		24	23.96	**98.6%**	1.4%	0.0%	0.7%	15.9%	**73.9%**	8.8%	0.7%	0.0%
	15	28	27.2	63.4%	36.6%	0.0%	0.0%	0.0%	21.0%	42.4%	26.8%	9.8%
		26	25.68	**96.7%**	3.3%	0.0%	2.2%	6.5%	68.5%	21.7%	1.1%	0.0%
		24	23.95	93.1%	6.9%	0.0%	6.9%	19.9%	**68.8%**	4.4%	0.0%	0.0%
Displacement (DV)	10	28	27.78	92.4%	7.6%	0.0%	0.0%	1.4%	42.4%	48.6%	5.4%	2.2%
		26	25.84	**98.2%**	1.8%	0.0%	0.0%	1.8%	**86.2%**	10.2%	1.8%	0.0%
		24	24.13	95.7%	4.3%	1.4%	2.9%	18.1%	76.4%	1.2%	0.0%	0.0%
	15	28	27.74	89.1%	10.9%	0.0%	0.0%	0.0%	37.7%	51.4%	10.5%	0.4%
		26	25.59	**96.7%**	3.3%	0.0%	2.2%	6.5%	68.5%	21.7%	1.1%	0.0%
		24	23.81	93.1%	6.9%	0.0%	6.9%	19.9%	**68.8%**	4.4%	0.0%	0.0%
Stratum (SV)	10	28	28.08	68.1%	31.9%	0.0%	0.0%	0.4%	16.3%	51.4%	29.7%	2.2%
		26	26.22	**100.0%**	0.0%	0.0%	0.0%	11.2%	**73.2%**	15.6%	0.0%	0.0%
		24	24.31	95.3%	4.7%	0.7%	4.0%	27.2%	67.8%	0.3%	0.0%	0.0%
	15	28	28.04	85.9%	14.1%	0.0%	0.0%	0.7%	45.3%	39.9%	12.7%	1.4%
		26	26.12	**95.0%**	5.0%	0.0%	0.0%	3.3%	**60.5%**	31.2%	3.6%	1.4%
		24	23.73	95.3%	4.7%	0.0%	4.7%	44.2%	51.1%	0.0%	0.0%	0.0%
Modified Stratum 1 (SV-1)	10	28	28.23	87.8%	12.2%	0.0%	0.0%	0.3%	50.3%	37.2%	10.1%	2.1%
		26	25.72	**97.0%**	3.0%	0.0%	1.0%	15.6%	**71.9%**	9.5%	1.0%	1.0%
		24	23.82	91.7%	8.3%	0.3%	8.0%	27.8%	62.8%	1.1%	0.0%	0.0%
	15	28	28.3	92.7%	7.3%	0.0%	0.0%	2.1%	54.5%	36.1%	4.5%	2.8%
		26	26.08	**93.8%**	6.2%	0.0%	1.7%	18.1%	**61.8%**	13.9%	4.2%	0.3%
		24	23.91	88.9%	11.1%	1.0%	10.1%	44.8%	41.7%	2.4%	0.0%	0.0%
Modified Stratum 2 (SV-2)	10	28	28.17	95.7%	4.3%	0.0%	0.0%	6.2%	66.3%	23.2%	4.3%	0.0%
		26	26.17	**96.7%**	3.3%	0.0%	2.9%	21.7%	**69.6%**	5.4%	0.4%	0.0%
		24	24.55	83.3%	16.7%	1.8%	14.9%	29.3%	54.0%	0.0%	0.0%	0.0%
	15	28	28.53	**96.0%**	4.0%	0.0%	1.1%	2.9%	64.1%	29.0%	2.5%	0.4%
		26	26.28	92.4%	7.6%	0.0%	7.6%	25.7%	60.5%	6.2%	0.0%	0.0%
		24	24.51	77.9%	22.1%	2.2%	19.9%	40.9%	36.6%	0.4%	0.0%	0.0%
Modified Stratum 3 (SV-3)	10	28	27.97	**97.3%**	2.7%	0.0%	1.1%	11.3%	**61.8%**	24.2%	1.6%	0.0%
		26	25.85	91.4%	8.6%	2.2%	6.5%	38.7%	51.6%	1.1%	0.0%	0.0%
		24	24.31	53.8%	46.2%	7.5%	38.7%	35.0%	16.7%	2.2%	0.0%	0.0%
	15	28	28.08	**100.0%**	0.0%	0.0%	0.0%	11.8%	**84.4%**	3.8%	0.0%	0.0%
		26	25.79	79.6%	20.4%	2.7%	17.7%	33.3%	46.2%	0.0%	0.0%	0.0%
		24	23.99	62.9%	37.1%	14.0 %	23.1%	45.7%	16.7%	0.5%	0.0%	0.0%

The highest percentage of subjects who voted neutral (0) for each ventilation mode during the test is presented in Table 3.5.

Table 3.5 Highest percentage to vote "neutral" & "acceptable" in each ventilation mode with the correspondence airflow supply and average room temperature

Ventilation mode	**Thermal comfort level#**		**Test chamber condition**	
	Highest percentage to vote "acceptable"	**Highest percentage to vote "neutral"**	**Air flow supply**	**Average room temperature**
MV	98.6%	73.9%	10 ACH	24°C
	68.5%	96.7%	15 ACH	26°C
DV	98.2%	86.2%	10 ACH	26°C
	96.7%	68.5%	15 ACH	26°C
SV	100%	73.2%	10 ACH	26°C
	95.0%	60.5%	15 ACH	26°C
SV-1	97%	71.9%	10 ACH	26°C
	93.8%	61.8%	15 ACH	26°C
SV-2	96.4%	69.6%	10 ACH	26°C
	64.1%	96.0%	15 ACH	28°C
SV-3	61.8%	**97.3%**	10 ACH	28°C
	100%	84.4%	15 ACH	28°C

"Acceptable" indicates a vote of -1, 0, or +1 and "Neutral" a vote of 0.

Figures 3.8 to 3.13 give us a comparison of the acceptance percentages of the six ventilation modes from 24℃ to 28℃. Pre-set room temperatures for human subject tests are indicated on these figures in x-axis, whereas the Y-axis expresses the 7-point vote in percentages compared with two air flow supplies of 10 ACH and 15 ACH. The thin (in brown) and thick (in black) lines of each figure represent two air flow supplies of 10 ACH and 15 ACH, respectively.

In MV, as shown in Figure 3.8, the highest acceptance rate of neutral (0) vote is 73.9% at 10 ACH and 24℃, and the highest acceptable percentage is 98.6% at 24°C and 10 ACH, as shown in Table 3.4. This result is in line with the general temperature settings of most Hong Kong air-conditioned offices and classrooms under MV (Mui et al, 2007) and also approaches the neutral temperature of 24.6℃ at 10 ACH.

The acceptance rate reached the peak 25.5℃, i.e., 73.9%, and the satisfaction rate is found to decline after 25.5 ℃. This implies that humans feel uncomfortable when the temperature is above this point. It is obvious that the acceptance level cannot be improved by increasing the air flow supply. Contrary to this, increasing air flow reduces the acceptance rate. This demonstrates that the thermal comfort environment needs to consider not only the air flow rate, but also the air distribution strategy. This phenomenon is quite different from stratum ventilation.

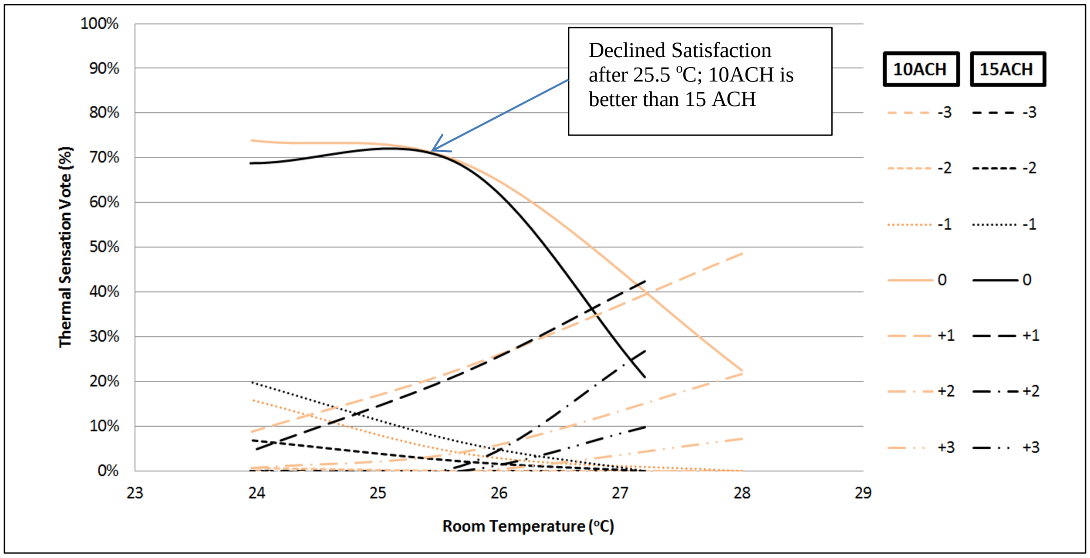


Figure 3.8 Distribution of thermal sensation vote (7-point scale) under mixing ventilation
(The highest percentages to vote "neutral" for 10 and 15 ACH are 73.9% at 24 ℃ and 96.7% at 26 ℃)

Figure 3.9 presents the trend of DV, the relationship between the acceptance percentage related to room temperature, and the influence of air flow are similar to MV that is the higher air flow supply cannot enhance the thermal comfort condition. But the highest acceptance rate of subject selecting neutral (0) under 10 ACH, which is 86.2% at 26℃ and 10ACH, is higher than the actual rate of 73.9% under MV at 24℃ and 10 ACH. Results show that 86.2% of subjects select neutral (0) at 26℃, which is higher than 76.4% at 24℃. In fact, the acceptance rate shown in the curve of neutral temperature at 10 ACH is better than 15 ACH. This implies that increasing air flow supply cannot enhance the thermal comfort condition under DV. This finding also matches the feature of low air velocity for displacement ventilation mode of providing a comfortable environment to the subjects.

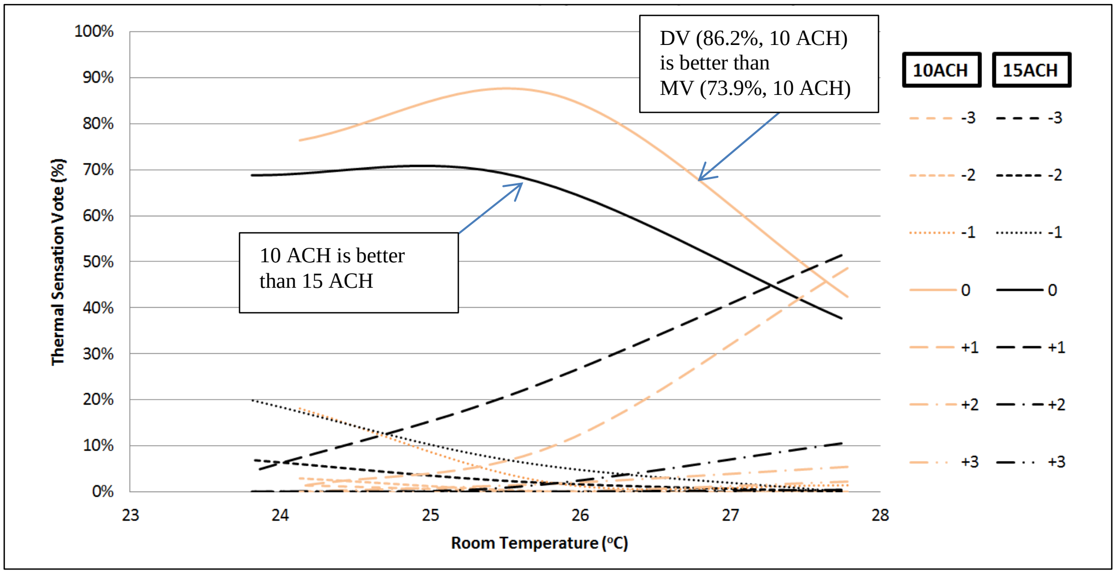


Figure 3.9 Distribution of thermal sensation vote (7-point scale) under displacement ventilation
(The highest percentages to vote "neutral" for 10 and 15 ACH are 86.2% at 26℃ and 68.5% at 26℃)

From Figure 3.10, 3.11, and 3.13, it indicates that better comfort conditions can be achieved by increasing the air flow supply at the room condition which should be higher than 26.8℃, 27.3℃, and 26.4 ℃ with respect to SV, SV-1, and SV-3 respectively. However, no significant improvement by increasing the air flow supply under MV, DV & SV-2 can be observed due to without any interception point of same thermal sensation vote with 10 ACH and 15 ACH shown in Figure 3.8, 3.9 & 3.12.

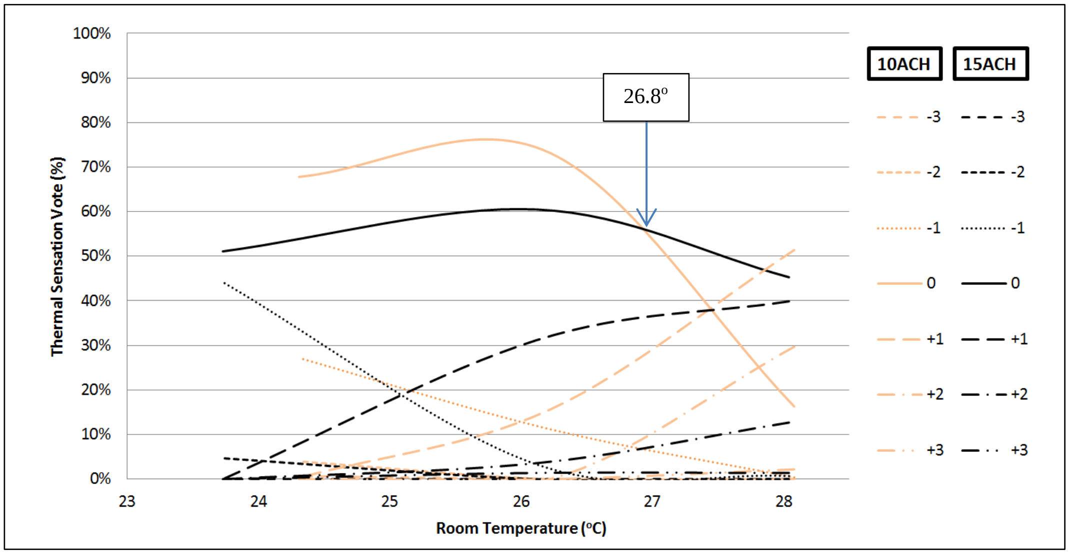


Figure 3.10 Distribution of thermal sensation vote (7-point scale) under stratum ventilation (The highest percentages to vote "neutral" for 10 and 15 ACH are 73.2% at 26℃ and 60.5% at 26℃)

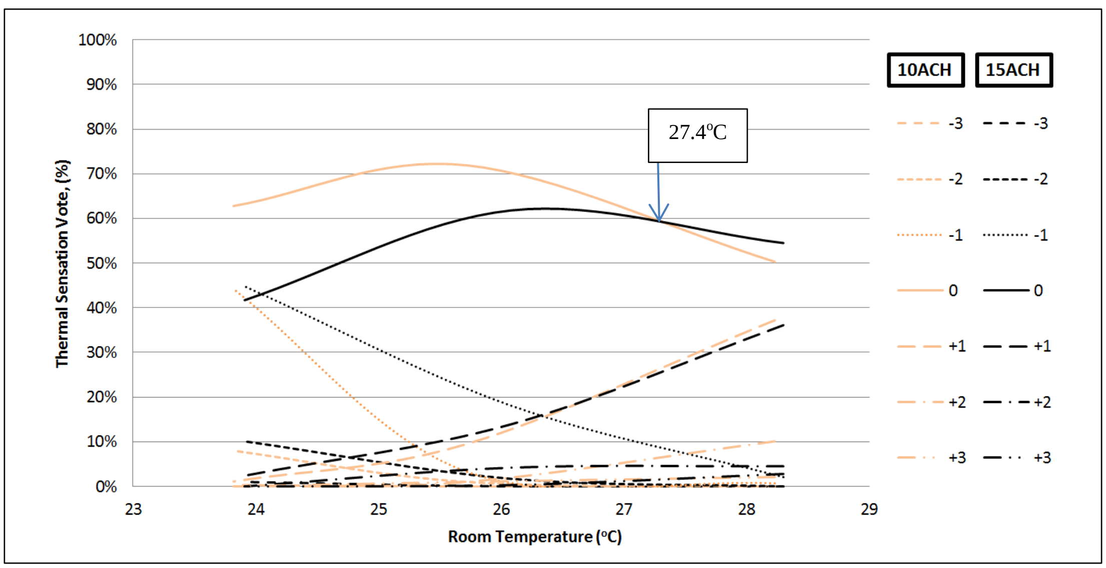


Figure3.11 Distribution of thermal sensation vote (7-point scale) under modified-stratum-1 ventilation *(The highest percentages to vote "neutral" for 10 and 15 ACH are 71.9% at 26℃ and 61.8% at 26℃)*

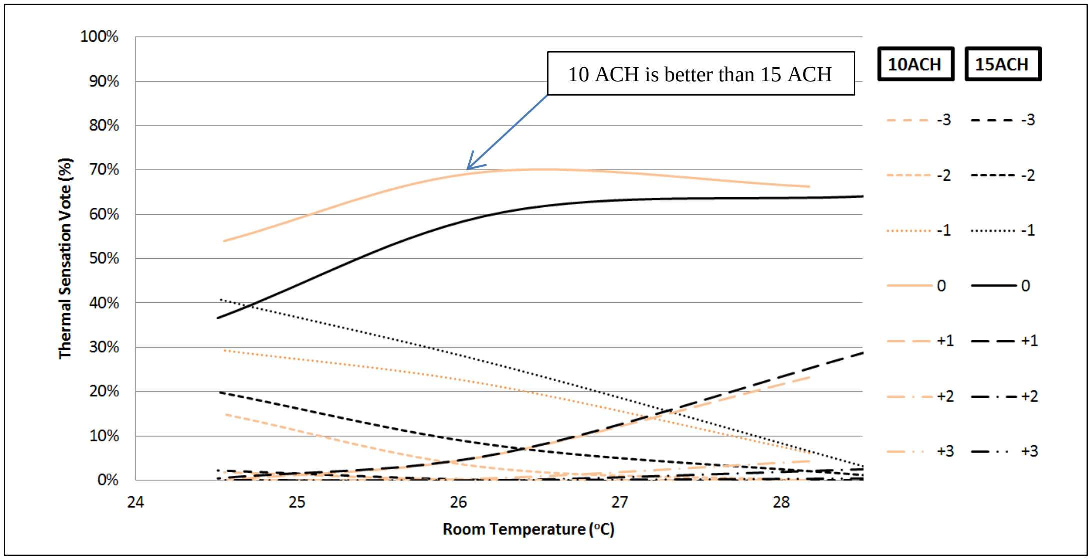


Figure 3.12 Distribution of thermal sensation vote (7-point scale) under modified-stratum-2 ventilation
(The highest percentages to vote "neutral" for 10 and 15 ACH are 69.6% at 26℃ and 96.0% at 28℃)

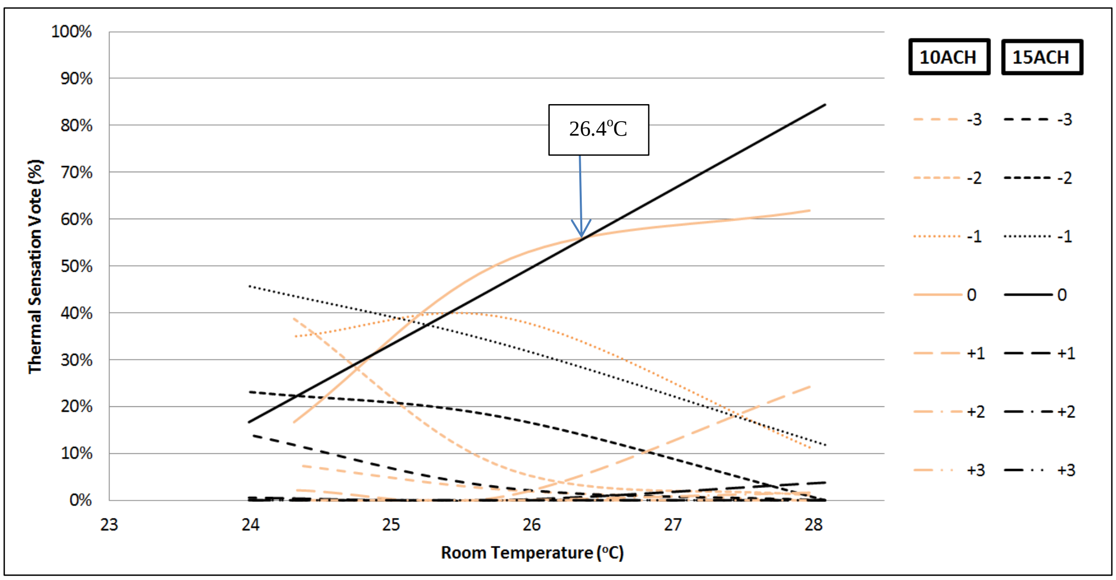


Figure 3.13 Distribution of thermal sensation vote (7-point scale) under modified- stratum-3 ventilation
(The highest percentages to vote "neutral" for 10 and 15 ACH are 97.3% at 28℃ and 84.4% at 28℃)

In comparing the six ventilation modes, the highest percentages to vote (0) of 97.3% is modified-stratum-3 mode at 28℃ under 10 ACH, respectively. In term of the trend of average acceptance percentage of neutral vote each vote, the interception points of two supply air flow only occurred at SV, SV-1 & SV&3 are found at 26.8, 27 & 26.4 ℃. It implies that increasing supply air flow cannot enhance the thermal comfort level at MV, DV & SV-2, but achievable in SV, SV-1 & SV-3 these critical temperatures of 26.8, 27 & 26.4 ℃. The better comfort conditions can be only achieved by increasing the air flow supply at the room condition which should be higher than 26.8℃, 27.3℃, and 26.4 ℃ with respect to stratum ventilation, modified-stratum-1, and modified-stratum-3 respectively. This implies that these may be the optimal design from a thermal comfort perspective. Of course, this is close to the estimated thermal neutral temperature and energy consumption aspect, as discussed in the following session.

3.4 Analysis of thermal neutral temperatures

The predicted mean vote (*PMV*) curve given in ISO standard 7730 is based on numerous comfort evaluations performed under controlled steady state conditions using thousands of randomly chosen subjects. One hundred percent satisfying a pre-set condition is statistically impossible. This *PMV* index can predict the mean values of scores from a large group of persons on a 7-point thermal sensation scale, based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment.

The mean value of all thermal sensation scores is given as a function of the testing time of the score. It seems that the mean votes are somewhat higher or lower at the time of the first, second, third, and fourth score; this is probably due to the subject's slight instability at the starting time of human comfort test. To ensure a steady-state condition, it is decided to consider only the last two scores in the final analysis out of a total of six scores in each session, i.e., the mean thermal sensation score taken from the last two scores is determined for each individual subject. A regression analysis is performed to determine the relationship between the population mean vote and room temperature. Thermal neutral temperature is defined in a thermal comfort context as the room temperature for which the predicted mean vote (*PMV*) of a sample occupant group is equal to zero; this depends on clothing, activity, personal constitution, age, etc.

In Table 3.6, for 48 male and female subjects under different conditions, the calculated regression equations for the mean thermal sensation vote are given as a function of the room temperature for the six ventilation modes. The room temperature corresponding to the neutral mean vote ($Y = 0$) is designated as the neutral temperature. The corresponding equations of the six ventilation modes at 10 & 15 ACH are shown in Figures 3.14 and 3.15.

Table 3.6 Regression equations for different groups of subjects with 48 males and females

Air Flow Supply	Ventilation Mode	Regression Equation	Neutral Temperature, ℃	R-Square
300 L/s, 10 ACH	Mixing	Y = 0.2515X - 6.1812	24.6	0.968
	Displacement	Y = 0.2074X – 5.2096	25.1	0.997
	Stratum	Y = 0.3755X –9.6056	25.6	0.9746
	Modified-stratum-1	Y = 0.2390X –6.2085	26.0	0.9965
	Modified-stratum-2	Y = 0.2841 – 7.7077	27.1	0.9936
	Modified-stratum-3	Y = 0.3542X - 9.6655	27.3	0.9754
450 L/s, 15 ACH	Mixing	Y = 0.3463X – 8.5806	24.8	0.997
	Displacement	Y = 0.2478X – 6.2809	25.3	0.995
	Stratum	Y = 0.2578X – 6.6473	25.8	0.9684
	Modified-stratum-1	Y = 0.2644X – 7.0317	26.6	0.9929
	Modified-stratum-2	Y = 0.3051X – 8.3712	27.4	0.9991
	Modified-stratum-3	Y = 0.2399X – 6.7022	27.9	0.9983

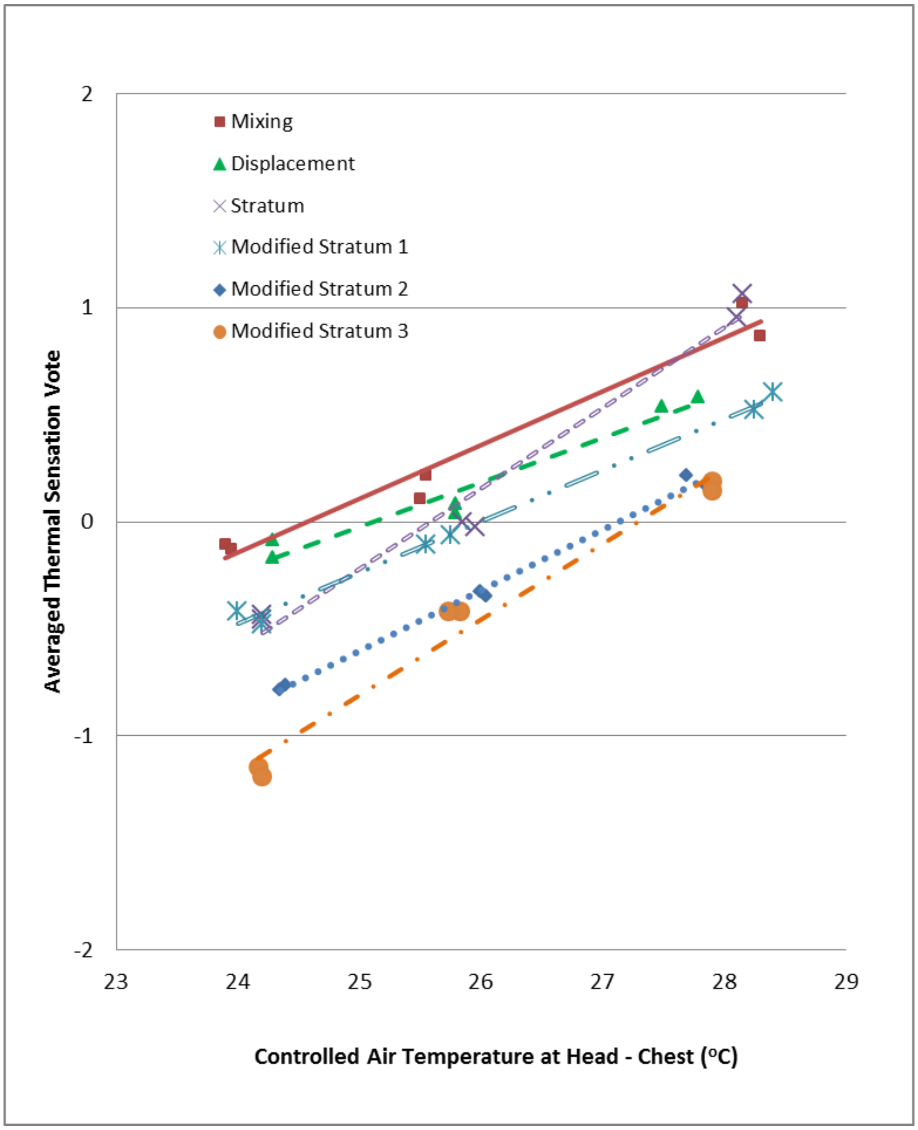


Fig 3.14 Relationship between actual mean vote and room temperature at 10 ACH

(Neutral temperatures of MV, DV, SV, SV-1, SV-2, and SV-3 are found to be 24.6 ℃, 25.1 ℃, 25.6 ℃, 26.0 ℃, 27.1 ℃, 27.3 ℃ at interception point of the correspondence regression equation with Y=0)

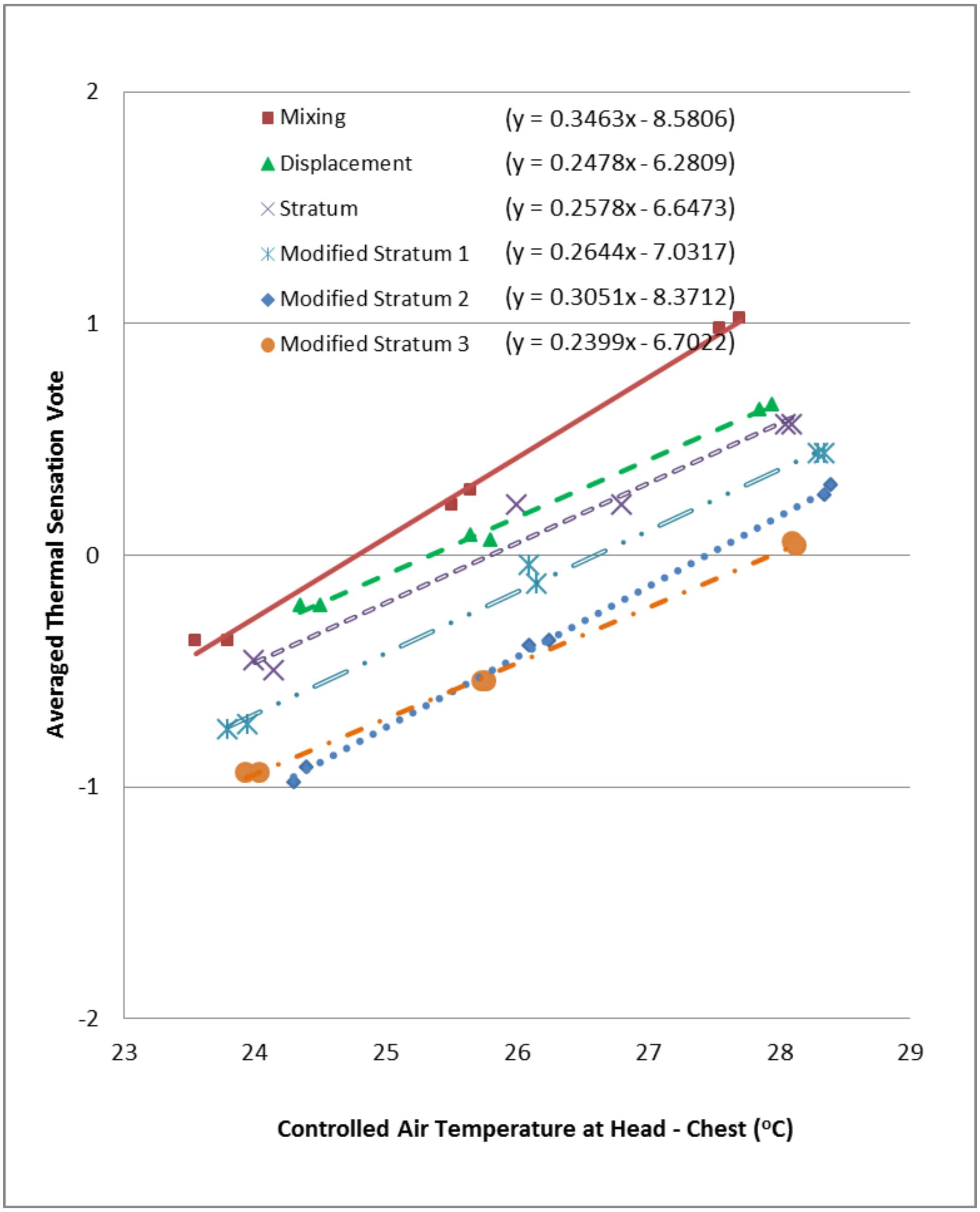


Fig 3.15 Relationship between actual mean vote and room temperature at 15 ACH

(Neutral temperatures of MV, DV, SV, SV-1, SV-2, and SV-3 are found to be 24.8 ℃, 25.3 ℃, 25.8 ℃, 26.6 ℃, 27.4 ℃, 27.9 ℃ at interception point of the correspondence regression equation with Y=0)

Table 3.6, Figures 3.15 & 3.16 indicated that the neutral temperatures of college students are found to be: 24.6℃ at 10 ACH and 24.8℃ at 15 ACH under mixing ventilation; and 25.1℃ at 10 ACH and 25.3℃ at 15 ACH under displacement ventilation. These findings for the conventional ventilation modes are similar to the previous reports of the neutral temperatures of 24.9 ℃ evaluated from 422 occupants' responses to the thermal environment under 0.1 m/s mean air speed in 61 air-conditioned office buildings in HKSAR (Mui al et. 2007), and of 25.4℃ estimated from 300 subjects under 0.1m/s to 0.2m/s air speed measured at 0.6 m above floor level (Chow et al. 2010). For stratum, modified-stratum-1, modified-stratum-2, and modified-stratum-3 ventilation modes, the neutral temperatures are found to be: 25.6℃, 26.0℃, 27.1℃, 27.3℃ at 10 ACH, respectively; and 25.8℃, 26.6℃, 27.4℃, 27.9℃ at 15 ACH, respectively. At a clothing level of 0.57 clo and a metabolic rate of 1 met for a sedentary working environment, increasing the air flow supply rate from 10 to 15 ACH results in a 0.2℃ rise in neutral temperature both for mixing ventilation and displacement ventilation modes. The corresponding amount is 0.2℃ to 0.6℃ for various stratum ventilation modes. In other words, by equally supplying 50% more air change from 10 to 15 ACH, stratum ventilation allows subjects to accept an additional 0.2℃ to 0.6℃, compared with an additional 0.2℃, under mixing ventilation and displacement ventilation modes. Therefore, air speed is a control variable that can enhance human thermal comfort. It is obvious that ventilation provides more of a thermal comfort enhancement effect at an elevated room temperature condition.

The relative humidity during these tests is found to be between 50% and 63%. Thus, this relative humidity will not affect the result of human comfort test due to the pervious finding on humidity are less sensitive than air temperature and air speed (Fong et al. 2010). Moreover, this humidity range is well within the ASHRAE comfort zone and should not affect the interoperation of collected data from the votes of human comfort test.

Among the six ventilation modes at 10 ACH, the neutral temperature of the modified-stratum-3 mode is about 2.7℃ (27.3℃ to 24.6℃) higher than that of the mixing mode, and 2.2℃ (27.3℃ to 25.1℃) higher than that of the displacement mode. At 15 ACH, the figures are 3.1℃ (27.9℃ to 24.8℃) and 2.6℃ (27.9℃ to 25.3℃), respectively. Therefore, thermal comfort level can be significantly enhanced with modified-stratum-3 ventilation in warm conditions. Compared with conventional air distribution modes, the higher neutral temperature of modified-stratum-3 is due to higher air velocity in the horizontal direction at the head-chest level.

Shown in Table 3.3b, the measured air speed at the center of the chamber at 1.1m above floor level only increases from the range of 0.15 m/s to 0.42 m/s to the range of 0.52 m/s to 0.76 m/s when the air flow supply rate is increased by 50% for stratum ventilation; in contrast, the values vary insignificantly for mixing ventilation and displacement ventilation. At the air speeds of 0.42 m/s and 0.76 m/s, the room temperature needs to be at least 24.4℃

and 25.4℃, respectively, to avoid draft according to Standard 55-2010. From the observation, the air speed of 0.76 m/s cannot provide a sufficient force to fly the paper which place in the table during these human comfort tests. This also explains why the thermal neutral temperatures found by human comfort tests are elevated under stratum ventilation.

3.5 Analysis of energy consumption

To remove heat from space by supplying a stream of cold air incurs energy consumption via two mechanisms: refrigeration power needed to produce the stream of cold air and power to circulate the cold air. The analysis in this section is referred to heat extraction from the estimated sensible space load of the environmental chamber to investigate the cooling energy serving by various ventilation systems.

The following assumptions and information are made:

1. Air flow rate per KW cooling: Fan power is a linear function of air flow rate. This would be the case where a statutory limit on specific fan power of 1.6 W per l/s for contact air volume system. It is based on clause 6.7.1, Code of Practice for Energy Efficiency of Building Services Installation 2012, EMSD. α=1.6 kW/m^3/s use as a criterion for this cooling energy estimation;
2. Coefficient of performance of chiller plant for mixing, displacement and stratum is depended on chilled water leaving temperature and cooling tower inlet temperature

as shown on Table 3.7. The *COP* ratio is retrieved from the data of "Samp C-DAT" of WaterCooledChiller in TRNSYS Manual. The fraction of Carnot *COP* of chiller, β for standard case is assumed as 1 and the other cases are shown in Table 3.7.

Table 3.7 Relationship between various water temperatures for different COP of chiller (Retrieved from TRNSYS, WaterCooledChiller, Samp C-DAT)

Ventilation method	COP Ratio	Chilled water leaving temperature, ℃	Cooling water inlet temperature, ℃
Standard case	1.000	7	30
Mixing	0.9011	7	35
Displacement	0.9191	8	35
Stratum	0.9393	9	35

3. Energy streams associated with refrigeration and distribution are equivalent, as would commonly be the case with electrically-driven equipment.
4. The specific volume, v_s [m^3/kg] extracted from Psychrometric chart at particular neutral temperature of each ventilation method are tabulated in Table 3.8.

Table 3.8 Specific volume of various ventilation methods

Ventilation Method	Enthalpy of room air [kJ/kg]	Specific volume [m^3/kg]
Air Circulation = 0.3 m^3/s (10 ACH)		
Mixing ventilation at 24.6°C	49.39	0.857
Displacement ventilation at 25.1°C	50.66	0.859
Stratum ventilation at 25.6°C	51.95	0.860
Modified-stratum-1 ventilation at 26.0°C	53.00	0.862
Modified-stratum-2 ventilation at 27.1°C	55.96	0.866
Modified-stratum-3 ventilation at 27.3°C	56.51	0.867
Air Circulation = 0.45 m^3/s (15 ACH)		
Mixing ventilation at 24.8°C	49.89	0.858
Displacement ventilation at 25.3°C	51.17	0.859
Stratum ventilation at 25.8°C	52.47	0.861
Modified-stratum-1 ventilation at 26.6°C	54.06	0.864
Modified-stratum-2 ventilation at 27.4°C	56.79	0.867
Modified-stratum-3 ventilation at 27.9°C	58.18	0.869

5. Design criteria of Internal space load:

- Total People Load = 17 persons (16 college students + 1 experimenter) x 115 W = 1955 W (Total of 115 W[a] = sensible of 70W + latent of 45W based on "Degree of Activity is seated, very light work", Table 3, Chapter 28, ASHRAE Fundamentals Handbook).[a] Total heat of 115 W - adjusted heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85% of that for an adult male, and that the gain from a child is 75% of that for an adult male.

- The breakdown of total Sensible load, Qs [kW] of 2.686 is shown in Table 3.9.

Table 3.9 Internal heat sources

Occupants	Equipment (projector, lamps & computer)	Total
1190 W (17 persons × 70[1])	1496 W (220 + 1176 + 100)	2686 W

Remark [1] for Table 3.9: People Load is based on "Degree of Activity is seated, very light work", Table 3, Chapter 28, ASHRAE Fundamentals Handbook)

- Fraction of Carnot of COP of Chiller, β
- Specific volume, vs [m^3/kg] read from Psychrometric chart
- Return air temperature is assumed equal to the indoor temperature for this cooling energy estimation
- Exhaust air temperature is equal to neutral temperature plus 2℃ for this case.

6. Basic equations for fan power and cooling energy estimation are shown as below:

$$P_f = \propto v \qquad \text{Eq. (3.1)}$$

$$Q = vC_v(T_i - T_s) \qquad \text{Eq. (3.2)}$$

$$P_c = \frac{Q}{\beta}\frac{(T_o - T_s)}{T_s} \qquad \text{Eq. (3.3)}$$

By substitution the above information and assumption into above Eq. (3.1) & (3.2) to estimate the fan power consumption, the results are shown in Table 3.10a and then adding with the required refrigeration power by Eq. (3.3) to come up the required total cooling sensible power are tabulated in Table 3.10b.

Table 3.10a Fan power consumption with various ventilation methods

Ventilation Mode	Mass flow rate, $m = V / v_s$ [kg/s]	Room Temperature, Tr [^oC]	Exhaust Temperature, Te [^oC]	Supply air Temperature, $Ts = Te - Qs / m$ [^oC]	Fan Power, $Pf = \alpha\ V$ [kW]
Air Circulation = 0.3 m^3/s					
Mixing ventilation	0.3501	24.6	26.6	19.24	0.48
Displacement ventilation	0.3492	25.1	27.1	19.72	0.48
Stratum ventilation	0.3488	25.6	27.6	20.21	0.48
Modified-Stratum-1 ventilation	0.3480	26	28	20.60	0.48
Modified-Stratum-2 ventilation	0.3464	27.1	29.1	21.66	0.48
Modified-Stratum-3 ventilation	0.3460	27.3	29.3	21.85	0.48
Air Circulation = 0.45 m^3/s					
Mixing ventilation	0.5245	24.8	26.8	21.89	0.72
Displacement ventilation	0.5239	25.3	27.3	22.38	0.72
Stratum ventilation	0.5226	25.8	27.8	22.87	0.72
Modified-Stratum-1 ventilation	0.5208	26.6	28.6	23.65	0.72
Modified-Stratum-2 ventilation	0.5190	27.4	29.4	24.44	0.72
Modified-Stratum-3 ventilation	0.5178	27.9	29.9	24.92	0.72

Table 3.10b Cooling sensible energy for various ventilation methods

Ventilation Mode	Refrigeration Power, Pc = Qs (To-Ts) / (ßx Ts) [kW]	Total Power, Pc +Pf [kW]	% difference
Air Circulation = 0.3 m^3/s			
Mixing ventilation	1.57	2.05	Base
Displacement ventilation	1.49	1.97	-4.08%
Stratum ventilation	1.41	1.89	-8.56%
Modified-Stratum-1 ventilation	1.37	1.85	-10.81%
Modified-Stratum-2 ventilation	1.27	1.75	-17.55%
Modified-Stratum-3 ventilation	1.25	1.73	-18.84%
Air Circulation = 0.45 m^3/s			
Mixing ventilation	1.30	2.02	Base
Displacement ventilation	1.22	1.94	-3.88%
Stratum ventilation	1.15	1.87	-8.02%
Modified-Stratum-1 ventilation	1.07	1.79	-12.66%
Modified-Stratum-2 ventilation	0.99	1.71	-17.69%
Modified-Stratum-3 ventilation	0.95	1.67	-21.04%

3.6 Concluding remarks

These findings were obtained from forty-eight subjects; each of the three subject groups (8 women plus 8 men, 16 men and 16 women) was in turn involved in a series of the 18 test sessions, 6 questionnaires per test session. Totally 5,184 questionnaires (i.e. 48 x 18 x 6) were collected. The highest acceptance percentage at different specific room temperatures with an air flow supply of 10 ACH (as shown in Table 3.4) was found to be 95.8%, which involved mixing at 24℃, displacement at 24℃, stratum at 24℃, modified-stratum-1 at 24 & 26℃, modified-stratum-2 at 26℃. When increasing the air flow supply to 15 ACH, the highest acceptance percentage of 95.8% was found at 28℃ under modified-stratum-2 stratum and modified-stratum-3 only.

The neutral temperatures of HKSAR people under the mode of mixing, displacement, stratum, modified-stratum-1, modified 2, and modified-stratum-3 are found to be: 24.6℃, 25.1℃, 25.6℃, 26.0℃, 27.1℃ and 27.3℃ at 10 ACH, respectively; and to be 24.8℃, 25.3℃, 26.6℃, 27.4℃, and 27.9℃ at 15 ACH, respectively. More energy is expected to be saved in the modified-stratum-3 mode due to the achievement of thermal comfort at elevated air temperatures. For the measurement of air temperature and velocity, the error analysis presented by Kline and McClintock (1953) is adopted. We assume that the variant R, which is the function of independent variants X_1, X_2, …, X_n, that is to say R = R (X_1, X_2,…X_n), the relative uncertainty ($\Delta R/R$) is given by the mathematic expression :

$\frac{\Delta R}{R} = [\sum_{i=1}^{n} (\frac{\Delta X_i}{X_i})^2]^{1/2}$. Both air temperature and velocity are measured directly. The measurement uncertainty of air velocity is found at 7.5%, and the measurement uncertainty for temperature is at 0.7%.

These results indicate that the modified-stratum-3 mode could provide a satisfactory thermal comfort level to rooms of temperatures up to 27.1℃ at 10 ACH and 27.9℃ at 15 ACH. Thus, this experimental result can provide a scientific basis for the feasibility of elevated room temperatures and suggests considerable potential for saving energy. Further research is needed to investigate the effects of the different variable parameters, such as type and number of air supply terminals, to determine the optimal configuration for stratum ventilation.

In comparison with mixing ventilation as presented in Table 3.10b, the cooling sensible energy saving at 10 and 15 ACH are in the range of 4.08 % to 18.84 % and 3.88 % to 21.04 % respectively. In term of highest percentage to vote “neutral” is modified-stratum-3 ventilation at 10 ACH, and the percentage of energy saving of displacement and stratum ventilation are 4.08 % and 18.84 % respectively, in comparison with mixing ventilation at 10 ACH are used in future cost effectiveness analysis in Chapter 5. The major items are tabulated in Table 3.11, which form a basic for energy cost estimation during user assessment phase for the modified-stratum-3 to quantify an actual benefit by life cycle costing approach.

Table 3.11 Energy data for cost effectiveness study

Ventilation mode under 10 air change per hour	Total power of refrigerant system and fan system [kW]	% difference
Mixing ventilation	0.3501	Base
Displacement ventilation	0.3492	-4.08%
Modified-Stratum-3 ventilation	0.3460	-18.84%

CHAPTER 4: MEASUREMENT AND ANALYSIS OF STRATUM VENTILATION

In a comparison of six air distribution strategies as mentioned in Chapter 3, the results indicate that modified-stratum-3 mode will not only provide a satisfactory thermal comfort level to rooms of temperatures up to 27.1℃ at 10 ACH and 27.9℃ at 15 ACH, but also less energy use due to reduced ventilation load. Less differentiate temperature between outdoor and room temperature, less the ventilation load is required. This finding is matching with the aim for creating a horizontal layer of supply air in occupants' head-chest level (breathing zone) accommodating this warm condition (Lin et al. 2005 & 2009).

The characteristic of Stratum ventilation is to supply cool and fresh air horizontally directly to the breathing zone of occupants with little attention to indoor air quality (IAQ) and thermal comfort of the upper zone (>1.5m from the floor if the occupants are sedentary). The lower zone (<0.8m from the floor) is not the target zone of IAQ either. This chapter focuses on positioning supply air terminals at the front-wall slightly above the height of occupants, and extracts air at the same height on the opposite wall. In the occupied zone served by stratum ventilation, air speed generally increases with height, whereas air temperature gradient is reverse with the lowest value at the head level (Tian et al. 2011). The strongest cooling effect is therefore formed at the head level where the cooling is needed most (Lin et al. 2011). It is named as modified-stratum-3 mode in the previous chapter, but stratum ventilation is used herewith.

This chapter further analyzes stratum ventilation to estimate the uniformity of thermal environments served by stratum ventilation. The confirmation of the uniform thermal environment served by stratum ventilation can have a number of implications for both quantifying and designing the thermal comfort in a stratum ventilated room. With respect to the former, it provides scientific basis for using existing thermal comfort parameters to describe stratum ventilation; and with respect to the latter, it enables, to large extent, adopting a methodology for air distribution design.

The relationships between thermal sensation and air distribution methods were analyzed subjectively under three ventilation methods: mixing ventilation, displacement ventilation and stratum ventilation (Fong et al. 2011). As mentioned in the previous chapter, it was found that stratum ventilation could achieve thermal comfort under room temperatures of up to 27℃, approximately 2.5℃ higher than that under mixing ventilation and 2.0℃ higher than that under displacement ventilation.

Comparing with conventional mixing and displacement ventilation, stratum ventilation performs the following characteristics: (i) higher supply air temperature due to shorter distance between the supply terminal(s) and occupants; (ii) elevated air movement at head-chest level, i.e. breathing zone, to most effectively offset the elevated air temperature effect at equal supply air volume; (iii) moderate reverse temperature gradient in the occupied zone (Tian et al. 2008). (iv) air temperature is lower at the head-chest level and higher at ankle level, which effectively cools the occupants' area surrounding their carotid arteries that need cooling most (Brown and Williams 1982, Cohen et al. 1989, Nunneley et al. 1982, Nakamura et al. 2008); (v) improved inhaled air quality because fresh air is delivered directly to the breathing zone; (vi) a quasi-stagnant zone is formed between the breathing zone and the floor. Reasonably controlling temperature in this zone helps to resolve the problem of "cold ankle" in displacement ventilation, but also to save energy by avoiding over-cooling of the lower zone; (vii) higher performance coefficient (COP) for the associate refrigerating plant due to higher supply air temperature.

For stratum ventilation, non-isothermal air jets entrain some surrounding air to enter the breathing zone directly. There is a concern on the variations in air temperature and velocity relative to different positions in the occupied zone though such variations actually exist for almost all non-industrial air distributions. In this study, the data acquired from three experimental series are analyzed to evaluate the uniformity of air velocity and temperature, and also thermal sensation of the subjects.

4.1 Experimental design, setup and procedure

Three human test series are conducted to evaluate the uniformity of thermal environments in a stratum ventilated chamber with 8.8m (L) × 5.1m (W) × 2.4m (H). Totally nineteen conditions are generated by adjusting room temperature, supply airflow rate and supply terminal type. Due to huge amount of human comfort tests thermal comfort and field measurement works involved, Test-series-1 is done by myself and Test-series-2 are done continuously by another two investigators during three separate summer times. In conducting this study, all experimental design, setup and procedure are similar to the previous Chapter 2. All room temperatures mentioned in Table 1a, 1b and 1c are measured and represented by the central monitoring sensors positioned at 1.1m height above the floor in the geometrical center of this environmental chamber, as illustrated in Figure 4.1a and Figure 4.1b forTest-series-1,Test-series-2 andTest-series-3 respectively.

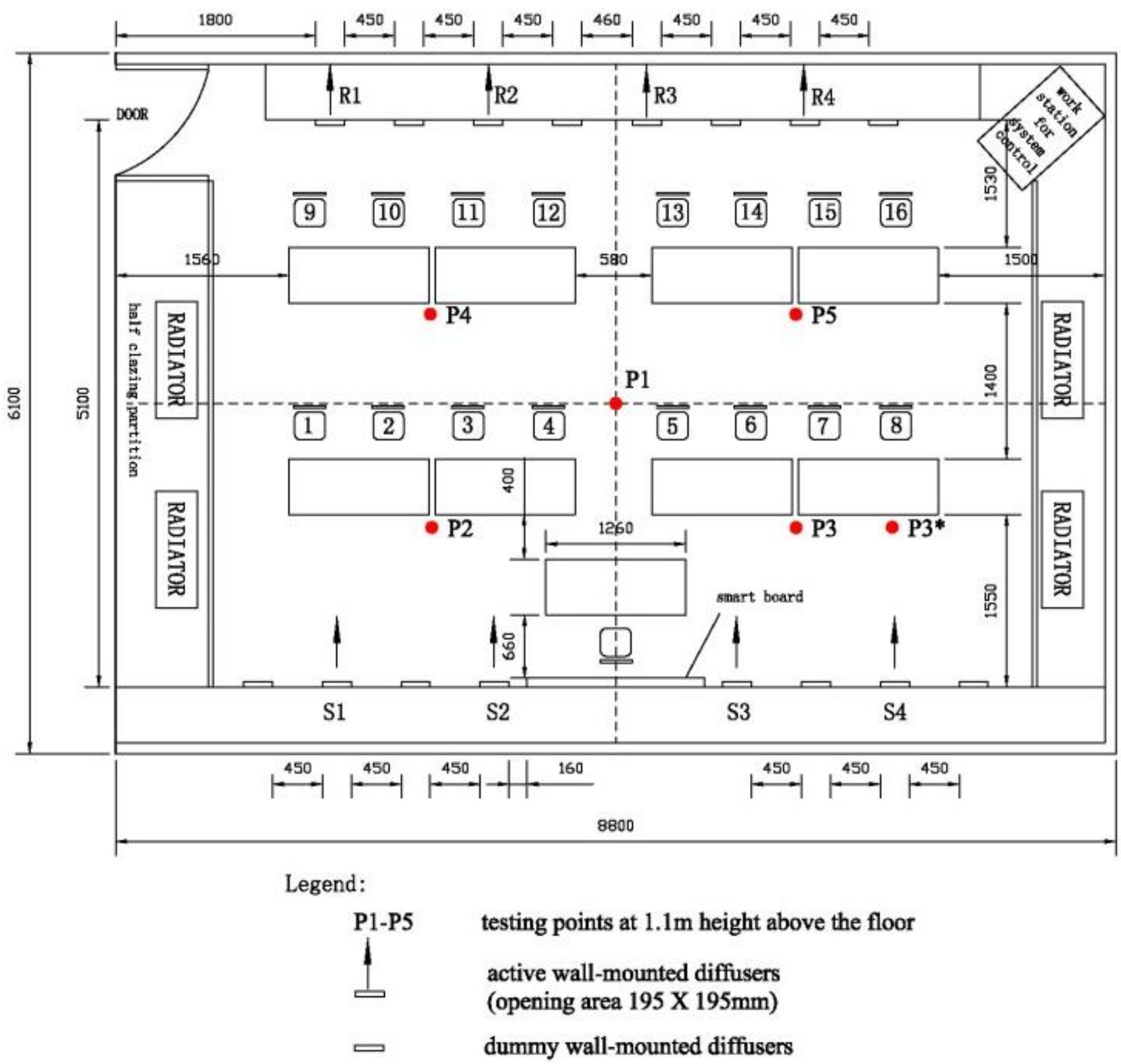


Figure 4.1a Experimental arrangement for Test-series-1 and 2

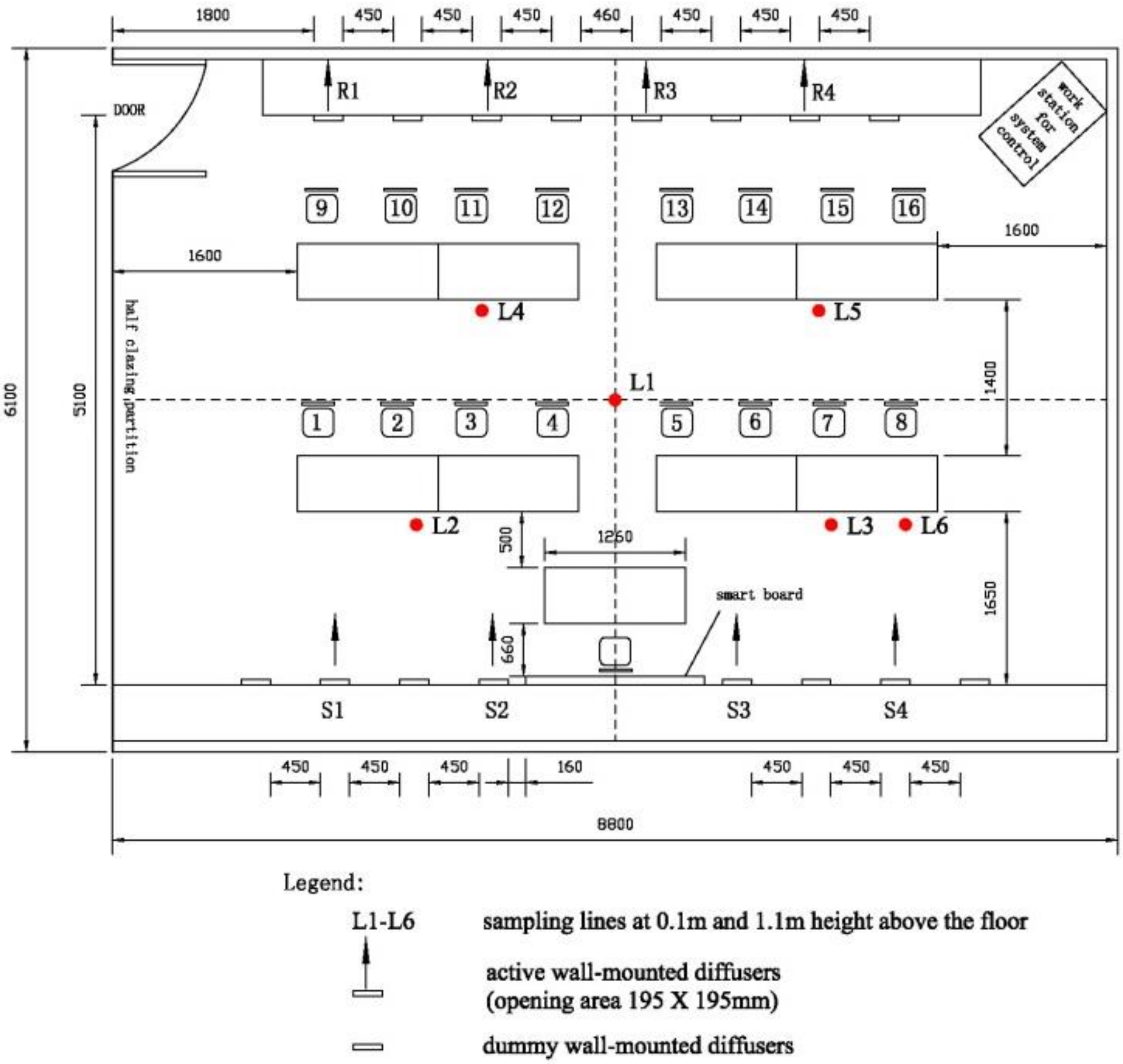


Figure 4.1b Experimental arrangement for Test-series-3

Three collected test series were analyzed in order to evaluate the uniformity of thermal environments served by stratum ventilation with a wide variety of experimental conditions. For Test-series-1, room temperature is varied from 24℃, 26℃ to 28℃ at a constant supply airflow rate of 10 air changes per hour (ACH) as shown on Table 4.1. From the result of

Test-series-1 in pervious Chapter 3, the neutral temperatures of stratum ventilation have been found at 27.1℃ at 10 ACH and 27.9℃ at 15 ACH. Two consequent testing room conditions by other two investigators are set to be 27℃ but make use of different range of air change rate, and type of diffusers to mutually support the uniformity of thermal environments during three separate data collected by different investigators. For Test-series-2, the supply airflow rate is varied from 7ACH to 17ACH at constant nominal room temperature of 27℃; twelve test sessions are tabulated in Table 4.1.

Table 4.1 Three Experimental Conditions

Experimental Series	Room temperature	Airflow rate	Type of diffuser
Test-series-1	28°C	10 ACH	Perforation
	26°C		
	24°C		
Test-series-2 (by investigator-1)	27°C	7ACH(0)	Perforation
		8ACH(0)	
		10ACH(0)	
		10ACH(2)	
		11ACH(0)	
		11ACH(2)	
		13ACH(2)	
		13ACH(3)	
		15ACH(3)	
		15ACH(4)	
		17ACH(3)	
		17ACH(4)	
Test-series-3 (by investigator-2)	27°C	10 ACH	Double
			Circular
			Square
			Perforation

Remark: ( ) represent number of radiators to be used during the human comfort test

In Test-series-3, the effect of various supply terminal types on the uniformity of air velocity and temperature has been examined. Four types of supply terminal are double deflection grille “Double”, circular diffuser “Circular”, square diffuser “Square” and perforated diffuser “Perforation”, as shown in Figure 4.2. The supply airflow rate and room temperature are kept constant at 10ACH and 27℃ respectively. Four test sessions for each type of diffusers have been conducted in Test-series-3 (Table 4.1).

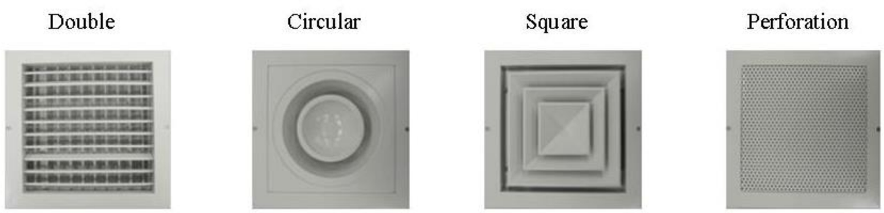


Figure 4.2 Supply terminal types

All three test series are conducted in the environmental chamber at City University of Hong Kong. The chamber is used to achieve a thermal environment with various air distribution strategies and the room air temperature in the chamber can be maintained with a precision of 0.5 °C. The environmental chamber is constructed to represent a classroom in sizes of 8.8 m (L) × 5.1 m (W) × 2.4 m (H) (Figure 4.1). The arrangements of furniture for Test-series-1 and Test-series-2 are the same (Figure 4.1a), while the arrangement of the working desks for Test-series-3 is slightly modified (Figure 4.1b). The installed air conditioning system consists of a ceiling mounted variable air volume type air handling unit, wall-mounted supply air diffusers for stratum ventilation and the associated motorized dampers and ductwork. The supply airflow rate could be varied by adjusting the operating frequency of the frequency conversion fan inverter. Air is supplied horizontally from four active wall-mounted diffusers (S1~S4) installed on the front wall at 1.3m level and returned via four wall-mounted return air terminals (R1~R4) on the rear wall at 1.3m the same height. In Test-series-1 and 2, perforated and double deflection diffusers are used separately. The indoor air relative humidity (RH) is controlled at 50 % to 65 %.

Each test session (a combination of settings) in the three test series involves sixteen subjects, who are seated by a working desk in four arrays and two rows. Each test session is designed to acquire the distributions of air temperature and velocity in the occupied zone, and the subjects' responses on their thermal sensation. To maintain the desired room temperature in Test-series-2, the certain number of radiator(s) is placed near the left and right walls in the room to provide the additional cooling load. An infra-red thermal tracer is used to check the room/furniture surface temperatures against the required room air temperature setting.

4.2 Air Diffusion performance index of stratum ventilation

Air Diffusion Performance Index (ADPI) is one of indicators to support the uniformity by a human comfort tests. ASHRAE standard 55-2010 has defined and estimated by the following equation:

$$\text{ADPI} = n/N \times 100\ (\%) \qquad (4.1)$$

Where,

ADPI = Air Diffusion Performance Index, %;
n = number of comfort points;
N = number of the total measuring points.

The number of the total measuring points "N" shall not less than the total concern area, m^2 divided by 9. And the comfort points "n" are determined with respect to the effective draft temperature (EDT) is between +1.5 and -1.0K, and local air speed (v_x) is lower than 0.35m/s. The difference in temperature between any point in occupied zone and the control condition, which is an effective draft temperature EDT, is defined as the following equation by Rydberg and Norback (1949) and later modified by Straub and Chen (1957), Straub et al. (1956) in the discussion of the paper by Koestel and Turve (1955):

$$\text{EDT} = (t_x - t_c) - 8(v_x - 0.15) \qquad (4.2)$$

Where,

EDT = effective draft temperature, K;
t_x = local airstream dry-bulb temperature, ℃;
t_c = average room dry-bulb temperature, ℃;
v_x = local airstream centerline speed, m/s.

According to the minimum achieved value of 80% ADPI, the uniformity of air velocity and temperature measured can be confirmed. Memarzadeh and Manning (2000) using ADPI for the uniformity and other indices, including *PMV* and *PPD* for thermal comfort, the local mean age of the air (LMAA) and the local air change index (LACI) for ventilation effectiveness, were employed to assess the performance of a ventilation system in a typical patient room.

The results showed that in the extreme winter conditions the most effective air change rate per hour (ACH) appeared to be 6, which produced the values of above 80% ADPI while also giving good values for LMAA and LACI; whereas for the typical winter cases a value of 4ACH would be adequate to ensure the uniformity of thermal environments. Wu et al. (2010) adopted numerical simulation to evaluate the effect of the diffuser arrangements on ADPI under 6 ACH and 12 ACH respectively. The diffuser arrangements would affect air diffusion and several adequate arrangements were found based on their ADPI values.

The uniformity of thermal environments with displacement ventilation was evaluated at five heights (0.1m, 0.6 m, 1.1 m, 1.7 m and 2.5 m) by Olesen et al. (1994). They reported that the ADPI values for most of the tests dropped below 80% even though generally the uniformity of the occupied zone (height ≤1.7 m) was superior to that of the entire space.

For stratum ventilation, non-isothermal air jets entrain some surrounding air to enter the breathing zone of occupants. Air temperature and velocity vary along the supply air paths. For the occupied zone, the largest variations in air temperature and velocity occur at head level because supply air jets are at this level (Tian et al. 2011). These variations represent the worst case in the occupied zone. If these variations at head level are confined within a certain range, so will the entire occupied zone. Variations in air velocity and temperature are significantly higher than those in the other parameters that affect thermal comfort, such as mean radiant temperature, relative humidity, etc. It is obvious that the higher the variations in air temperature and velocity are, the more difficult to achieve universal thermal comfort in a room. The uniformity of thermal environment under stratum ventilation therefore needs to be assessed. This part of study will mainly focus on the evaluation of the uniformity at the breathing zone of human subject at 1.1 m AFFL.

4.3 Experimental measurements

For three test series, both the objective measurements of air velocity and temperature distributions and the subjective evaluations of thermal sensations to thermal environments are carried out in accordance with the experimental setup as stated in previous Chapter 2.3 excepting one measuring sample at different location. Whereas, in Test-series-2, the measuring point (P3*) is intentionally changed to locate in the front of Seat Number 8 where it is just oriented at supply air terminal "S4". In Test-series-3, air velocity and temperature are measured at two planes. One is the ankle-level plane (0.1 m), and the other is the head-level plane (1.1m). Air speed and temperature of six locations (L1~L6) are measured (Figure 4.1a). For the location L1, air parameters are measured at the 1.1m level only.

4.3.1 Experimental procedure

All of participants in the three separate test series during three continuous summer times are university students of City University of Hong Kong, aged between 20 and 23 years old and either born and/or resided in HKSAR for more than ten years and the clo-value of the clothes and activity levels are limited to 0.57 clo and 1.0 met, respectively. The testing procedures and methodology were the same as those mentioned in previous Chapter 3.2 excepting an alternative as mentioned in the chapter.

4.3.1.1 Test-series-1

In Test-series-1, forty-eight human subjects (24 females and 24 males) were employed to participate in my research work. The anthropometric data of the subjects are shown in Table 3.1 of Chapter 3. Each test condition would be repeated three times with three different subject groups. The last two vote from six completed questionnaires collected for each session help to assess the quality of the final response, which was subsequently used in the analysis during each 30-minutes test session. The subjects voted their thermal sensations in 5 min intervals (i.e. six questionnaires times 5 min interval = 60 minutes or 0.5 hour in each session).

4.3.1.2 Test-series-2 (by other investigator-1)

Thirty-two subjects in two groups of 16 females and 16 males were employed to take part in theTest-series-2. The anthropometric data of the subjects are given in Table 4.2a. Each of the two subject groups in turn participated in a series of 12 test conditions as shown in Table 4.3. Each test condition was therefore repeated once one more time. The duration of each session was 1.5 hours. The subjects voted their thermal sensations in 20min intervals (i.e. six questionnaires times 15-min interval = 90 minutes or 1.5 hour in each session). Before the scheduled time of each session, the subjects were required to wait for 15min in the chamber for adapting to the thermal environment. During the tests, each subject was required to vote for his/her thermal sensation of ASHRAE 7-point scale in 15min intervals.

Table 4.2a Anthropometric data of subjects participating in Test-series-2 (by other investigator-1)

Gender	Statistic	Age	Height , H (cm)	Weight ,W (kg)	BMI (kg/m²)	BSA (m²)
Male	Mean	21.3	173.0	66.3	22.1	1.8
	S.D.	1.0	6.0	7.2	2.2	0.1
Female	Mean	20.9	160.0	51.6	20.0	1.5
	S.D.	0.9	6.0	6.8	2.0	0.1
Male + Female	Mean	21.1	167.0	59.0	21.1	1.6
	S.D.	1.0	9.0	10.1	2.3	0.2

* Body Mass Index (BMI) = W / H^2, W is weight in kg, H is height in m.
** Body Surface Area (BSA) = $(W^{0.425} \times 100\ H^{0.725}) \times 0.007184$, W is weight in kg, H is height in m.

4.3.1.3Test-series-3 (by other investigator 2)

Thirty-two university students were recruited to attend Test-series-3. The anthropometric data of the subjects are showed in Table 4.2b. They were equally separated into two groups, and each test group was required to experience the four experimental conditions as shown in Table 4.4. Thereby each experimental condition was implemented twice. Each test session lasted 2 hours. The subjects voted their thermal sensations in 20min intervals.

Table 4.2b Anthropometric data of subjects participating inTest-series-3 (by investigator-2)

Gender	Statistic	Age	Height , H (cm)	Weight ,W (kg)	BMI (kg/m^2)	BSA (m^2)
Male	Mean	20.8	174.8	63.1	20.6	1.8
	S.D.	0.9	7.5	13.1	3.5	0.2
Female	Mean	21.8	160.2	52.3	20.3	1.5
	S.D.	1.3	5.7	9.4	3.2	0.1
Male + Female	Mean	21.3	167.5	57.7	20.4	1.6
	S.D.	1.2	9.9	12.5	3.3	0.2

* Body Mass Index (BMI) = W / H^2, W is weight in kg, H is height in m.
** Body Surface Area (BSA) = ($W^{0.425}$ × 100 $H^{0.725}$) × 0.007184, W is weight in kg, H is height in m.

4.3.2 Actual experimental conditions

In Test-series-1, the room temperature was altered while the airflow rate was constant at 10ACH. Reversely in Test-series-2, the supply airflow rate was varied at constant room temperature of 27℃. In Test-series-3, the supply terminal type was changed while both supply parameters were kept constant. The thermal environments in the three test series cover a considerably wide range of conditions.

Tables 4.1, 4.3a, 4.3b and 4.3c show the summary of nominal and actual experimental conditions for Test-series-1, Test-series-2 and Test-series-3 respectively. It is found that actual room temperatures are close to nominal room temperature, which indicates that the actual experimental conditions are well controlled. The actual indoor relative humidity is measured within the range of 51% to 66%, which is well within the ASHRAE thermal comfort zone (ASHRAE 55-2010).

Table 4.3a Parameter combinations of Test-series-1 at 10 ACH

Session (no. of participant and gender)	Nominal room temperature (°C)	Actual room temperature (°C)	Supply face velocity (m/s)	Supply air temperature (°C)	Measured RH (%)
1 (16 males)	24	24±0.5^	1.73	15.5	51-57
2 (16 females)	26	26±0.5		16.9	52-57
3 (8 males + 8 females)	28	28±0.5		21.1	52-63

^ Standard Deviation (S.D.)

Table 4.3b Parameter combinations of Test-series-2 at nominal room temperature of 27 °C by other investigator-1

Session (no. of participant and gender)	Air change per hour	Actual room temperature (°C)	Supply face velocity (m/s)	Supply air temperature (°C)	Measured RH (%)
1m (16 males) and 1f 16 females)	7ACH(0)*	27.4±0.2	1.22	26.3	53-59
2m (16 males) and 2f (16 females)	8ACH(0) *	27.0±0.3	1.62	25.8	51-55
3m (16 males) and 3f (16 females)	10ACH(0) *	27.4±0.2	1.91	26.5	55-62
4m (16 males) and 4f (16 females)	10ACH(2) *	26.6±0.2		20.5	52-57
5m (16 males) and 5f (16 females)	11ACH(0) *	27.2±0.2	2.23	24.9	52-59
6m (16 males) and 6f (16 females)	11ACH(2) *	26.7±0.3		21.0	56-64
7m (16 males) and 7f (16 females)	13ACH(2) *	27.2±0.3	2.56	22.1	59-66
8m (16 males) and 8f (16 females)	13ACH(3) *	27.0±0.3		20.4	57-62
9m (16 males) and 9f (16 females)	15ACH(3) *	27.0±0.3	2.81	21.6	55-60
10m (16 males) and 10f (16 females)	15ACH(4) *	26.6±0.3		18.9	52-58
11m (16 males) and 11f (16 females)	17ACH(3) *	27.3±0.2	2.83	21.3	56-61
12m (16 males) and 12f (16 females)	17ACH(4) *	26.7±0.2		18.2	53-60

( )* The number of radiator(s) used to maintain the nominal room temperature

Table 4.3c Parameter combinations of Test-series-3 at nominal room temperature of 27 °C and 10ACH by other investigator-2

Session (no. of participant)	Supply terminal type	Actual room temperature (°C)	Supply face velocity (m/s)	Supply air temperature (°C)	Measured RH (%)
1a(16 subjects) and 1b (16 subjects)	Double	26.9±0.1	2.08	20.5-21.5	53
2a(16 subjects) and 2b (16 subjects)	Circular	26.8±0.1	4.49		52
3a(16 subjects) and 3b (16 subjects)	Square	26.9±0.2	4.46		52
4a(16 subjects) and 4b (16 subjects)	Perforation	26.7±0.1	3.64		51

4.3.3 Local air temperature and air velocity distributions

The measured local air temperature and velocity distributions for the three test series are illustrated in Figures 4.3 to Figure 4.5. In comparing with the distributions of air velocity, air temperature distributions at each measuring points, these patterns are uniform. Figures 4.3a & 4.3b depicted the distributions of room temperature and air velocity at breathing zone of human subject at 1.1 m AFFL for the three pre-set room temperatures of Test-series-1. At supply airflow rate of 10ACH, air velocities of four measuring points are lower than 0.35m/s. Only one measuring point "P5" is higher than 0.35m/s, which occurs in the session of room temperature 26℃. The locations of P1, P2, P3, P3*, P4 and P5 are illustrated in Figure 4.1a.

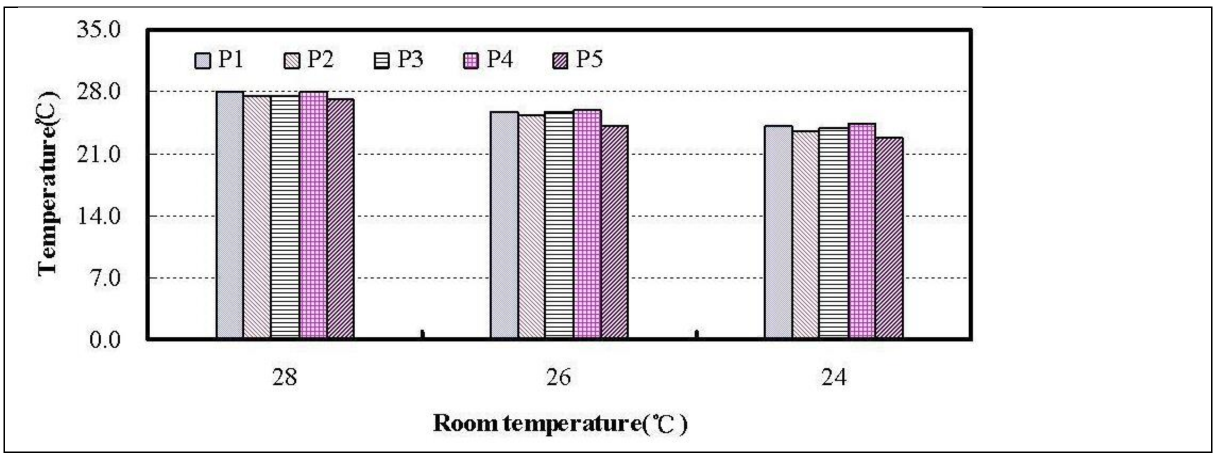


Figure 4.3a Local air temperature at 1.1m level in Test-series-1
(P1, P2 and P3 at 1st Row; P4 and P5 at 2nd Row)

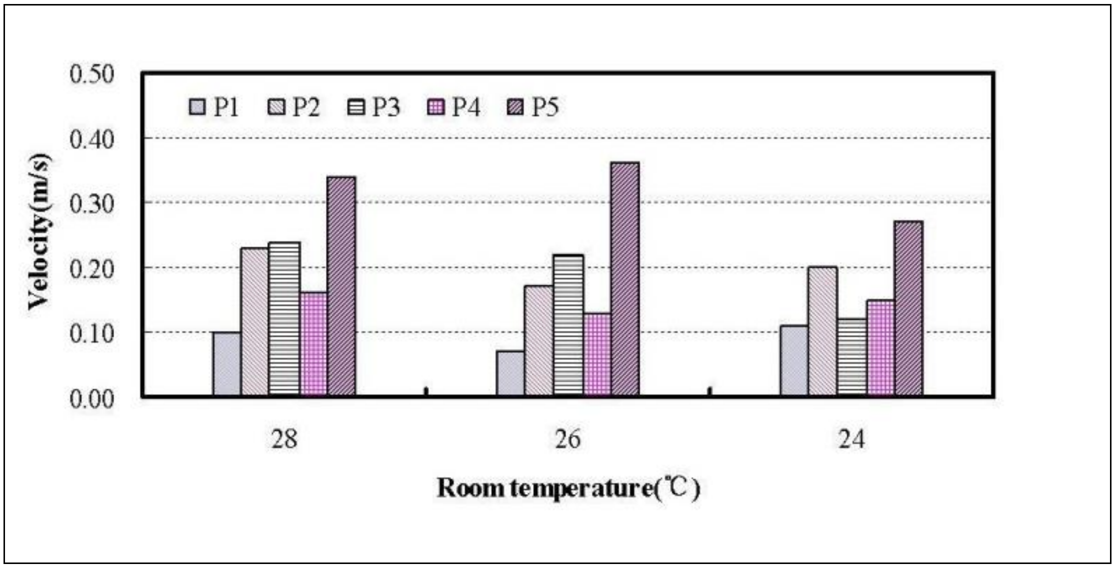


Figure 4.3b Local air velocity distributions at 1.1m level in Test-series-1
(P1, P2 and P3 at 1st Row; P4 and P5 at 2nd Row)

For the twelve experimental sessions of Test-series-2 as shown in Figures 4.4a & 4.4b, local air temperature and velocity under the room temperature of 27℃ are also measured at 1.1m level above the floor. It is found that local air temperature and velocity of these measuring points are not significantly affected by supply airflow rate, except for the measuring point of P3*. With the elevation of supply airflow rate, the velocity of P3* is obviously increased, and air temperature is correspondently lower than that of the other measuring points because P3* is within the air jet of the supply terminal of S4. Whereas, for the other four measuring points, the measured local air temperatures are similar and close to each other, and the air velocities are low. Particularly, when supply airflow rate is up to 17ACH, the velocity for some measuring points is elevated to higher than 0.35m/s, such as the measuring point of P1. The blanket of value as shown at the end of each ACH in x-axis of Figure 4.4a & 4.4.b represents the number of radiator(s) used to maintain the nominal room temperature.

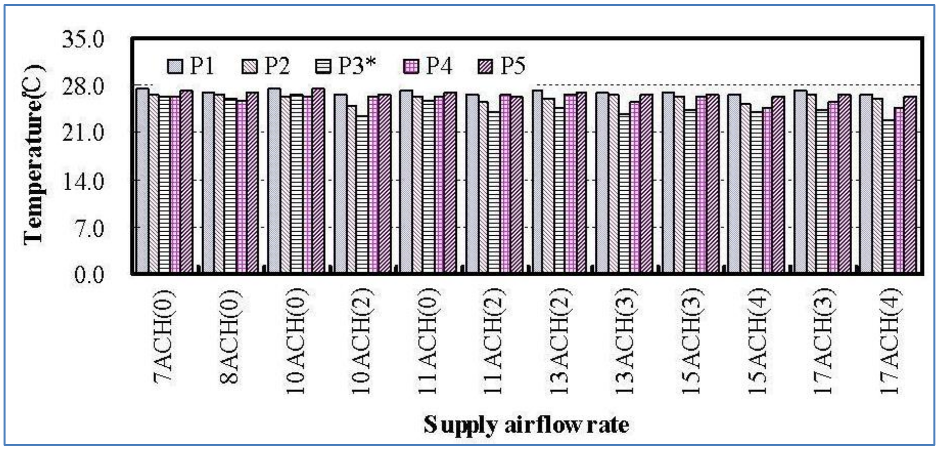


Figure 4.4a Local air temperature at 1.1m level in Test-series-2
(P1, P2 and P3* at 1st Row; P4 and P5 at 2nd Row)

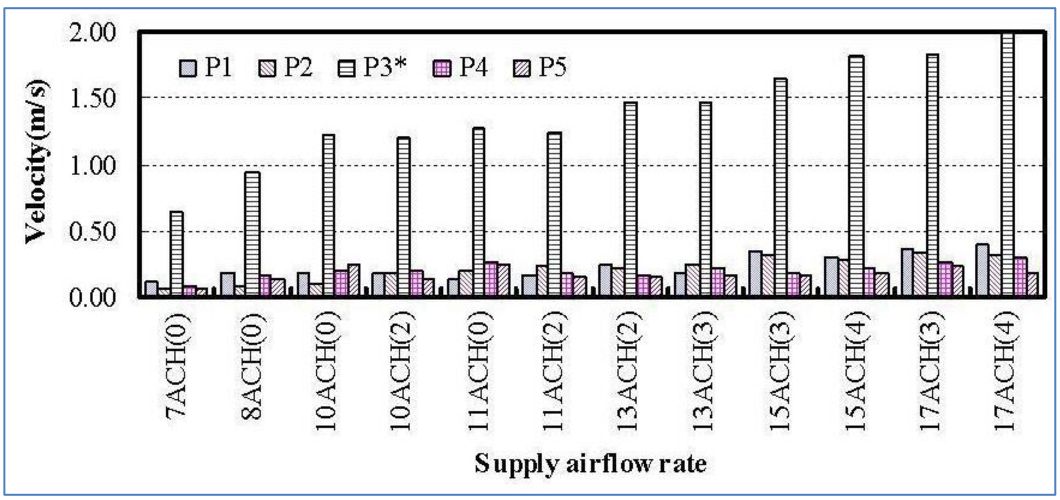


Figure 4.4b Local velocity distributions at 1.1m level in Test-series-2
(P1, P2 and P3* at 1st Row; P4 and P5 at 2nd Row)

For Test-series-3, local air temperature and velocity of two levels are measured. Those are measured on ankle level and head level of human subjects at 0.1m and 1.1m AFFL respectively. These profiles are showed in Figure 4.5. The results indicated that under the condition of constant room temperature 27℃ and supply air volume of 10ACH, supply terminal types have a more significant impact on local air velocity distributions than on local air temperature. For the ankle level at 0.1m AFFL, air velocities are all low for the four supply terminal types. At the breathing zone of human subject at 1.1 m AFFL, air velocities are generally higher, especially for Point L6, which is the nearest point and directly facing an air supply. The location of Point L1, L2, L3, L4, L5 and L6 are shown in Figure 4.1b. It is ascribed to the effect of the supply air jet and illustrated in Figures 4.5a & 4.5b. In comparison with local air temperature and velocity at different heights at 1.1m height in Figures 4.5c and 4.5d, it found that the variations of air temperature and velocity at 1.1m height are higher than that at 0.1m level. This is consistent with the finding of Tian et al. (2011).

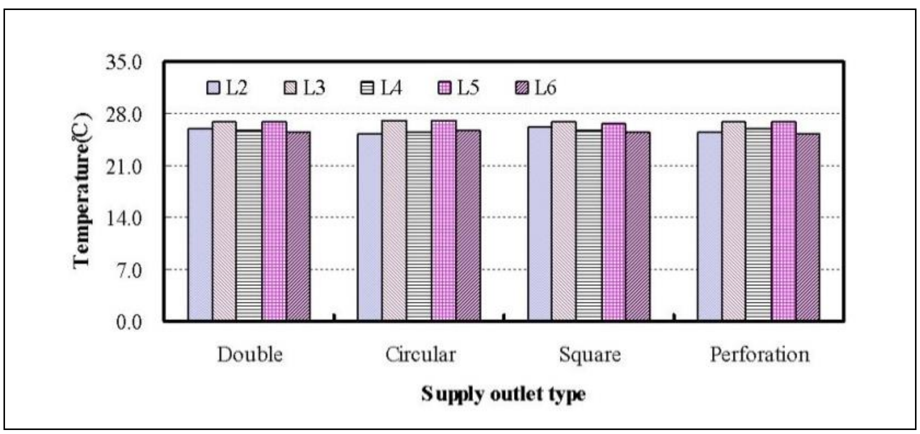


Figure 4.5a Local air temperature distributions at ankle level (0.1m AFFL) in Test-series-3 (L1 at center, L2, L3 and L6 at 1st row; L4 and L5 at 2nd row)

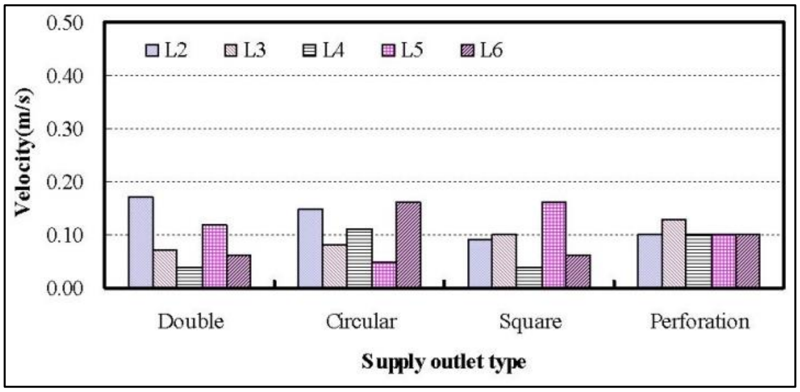


Figure 4.5b Local air velocity distributions at ankle level (0.1m AFFL) in Test-series-3
(L1 at center, L2, L3 and L6 at 1st row; L4 and L5 at 2nd row)

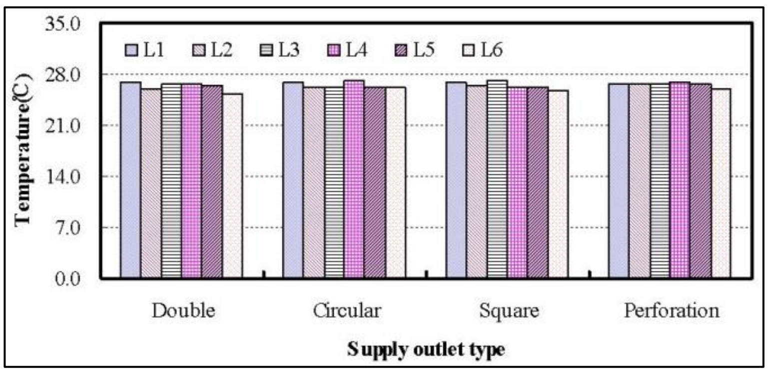


Figure 4.5c Local air temperature distributions at breathing zone (1.1m AFFL) in Test-series-3
(L1 at center, L2, L3 and L6 at 1st row; L4 and L5 at 2nd row)

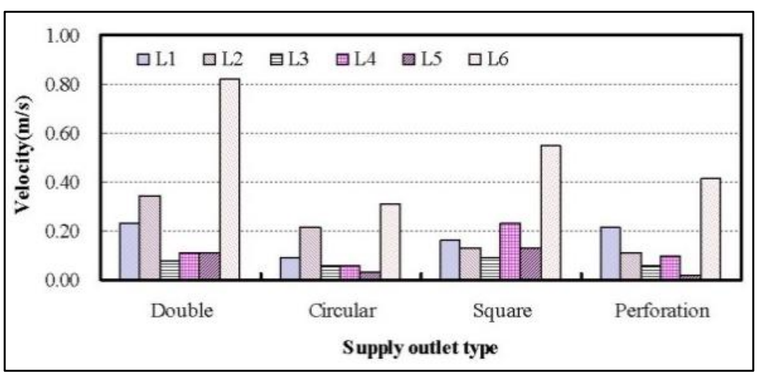


Figure 4.5d Local air velocity distributions at breathing zone (1.1m AFFL) in Test-series-3
(L1 at center, L2, L3 and L6 at 1st row; L4 and L5 at 2nd row)

As aforementioned, the uniformity of thermal comfort condition of a room can be evaluated by the air diffusion performance index (ADPI). The higher ADPI values, the more uniform thermal environment can be confirmed. To ensure the reasonable uniformity of the local air temperature and velocity conditions for universal thermal comfort, at least 80% ADPI value is necessary (ASHRAE 2013). The ADPI values for every test session of all the three test series are determined based on the measured values of local air temperature and velocity and summarized in Table 4.4.

Table 4.4 ADPI values for all test sessions

Experimental Series	Experimental session	ADPI (%)	Points of discomfort
Test-series-1	28°C	80	P5
	26°C	60	P1, P5
	24°C	80	P5
Test-series-2 (by investigator-1)	7ACH(0)	80	P3*
	8ACH(0)	80	P3*
	10ACH(0)	80	P3*
	10ACH(2)	80	P3*
	11ACH(0)	80	P3*
	11ACH(2)	80	P3*
	13ACH(2)	80	P3*
	13ACH(3)	80	P3*
	15ACH(3)	80	P3*
	15ACH(4)	80	P3*
	17ACH(3)	40	P1,P3*,P4
	17ACH(4)	40	P1,P3*,P4
Test-series-3 (by investigator-2)	Double	82	L2 (1.1m), L6 (1.1m)
	Circular	82	L2 (0.1m), L6 (1.1m)
	Square	91	L6 (1.1m)
	Perforation	91	L6 (1.1m)

Remark: Room temperature of Test-series-1 and Test-series-2 is equally pre-set to 27°C.

For most cases in Table 4.4, the ADPI values are equal to or greater than 80%. This reveals, through these objective experiments, that the occupied zone of the stratum ventilated room is a uniform thermal environment. In the excessive supply air velocity of 17ACH, it found the uniformity in the occupied zone was degraded

4.3.4 Thermal sensation

In this study, subjective thermal sensation is also evaluated by the last votes. Figures 4.6, 4.7 & 4.8 shows mean thermal sensation of subjects for various sessions of Test-series-1, 2 and 3 respectively. Figure 4.6 illustrated mean thermal sensations of different room temperatures in Test-series-1. Room temperature has a great impact on thermal sensation. With the reduction of room temperature from 28℃ to 24℃, mean thermal sensation spans from +0.15, -0.25 to -0.92, respectively. Evidently, neutral thermal sensation should occur between room temperature of 28℃ and 26℃. This result was reported in previous Chapter 3 and his paper (Fong et al. 2011). They concluded that the thermal neutral temperature of stratum ventilation was slightly over 27℃. Under room temperature of 24℃, the slightly cool sensation of the subjects was found. This reveals that room temperature of 24℃ may not be suitable for stratum ventilation, whereas this temperature actually is good for mixing ventilation (Mui and Wong 2007).

In Test-series-2, Figure 4.7 shows the profiles of mean thermal sensations under different supply airflow rates while the room temperature is kept at 27℃. Even though supply airflow rate is significantly increased from 7ACH to 17ACH, the thermal sensations only exhibited small variations. This is because of a strong preference for more air movement under warm conditions (Arens et al. 2009). In Figure 4.8 for Test-series-3, the supply terminal type shows a certain impact on thermal sensation. This is due to the flow pattern formed by various types of supply terminals.

For various thermal environments formed in Test-series-1, 2 and 3, the standard deviations of thermal sensations of subjects are also given in Figures 4.6 to 4.8 respectively. The standard deviations are between 0.60 and 0.96 for Test-series-1, 0.44 and 0.67 for Test-series-2, and 0.78 and 0.95 for Test-series-3. Yu et al. (2003) evaluated thermal sensations for four mixing ventilation cases at room temperatures of 22℃, 23℃, 24℃ and 26℃.. The standard deviations of the actual thermal sensation votes for the four cases were 0.87, 1.02, 0.87 and 1.03 respectively. Balazova et al. (2008) conducted laboratory experiments on thermal sensation in six open-plan office environments with mixing ventilation. Two office temperatures level at 23℃ and 28℃ were used. They found that the average thermal sensations were significantly different at two office temperatures. Standard deviations of thermal sensations for all tested cases were varied between 0.70 and 0.89. Based on the large data set of this investigation, the standard deviations of thermal sensations of the subjects under stratum ventilation are comparable to those for rooms with mixing ventilation found in the literature. The latter is widely regarded as a uniform thermal environment. Data analysis has been conducted to assess the difference of thermal sensations between the subjects seated in the first row (Seat Number 1 to 8) and in the second row (Seat Number 9 to 16) for all cases in the three test series. It is found that for most cases studied, the differences in thermal sensation between the first row and the second row subjects are statistically insignificant (P > 0.05).

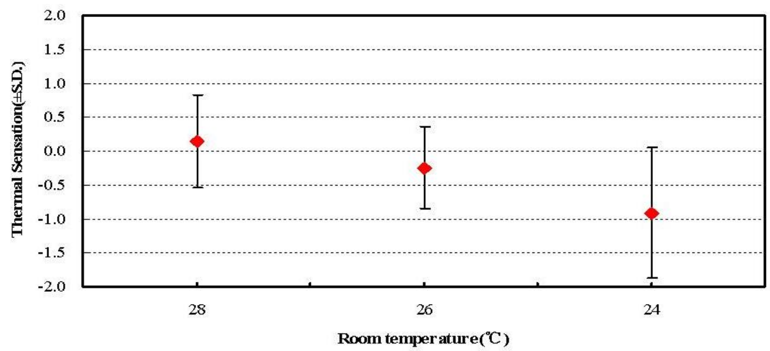


Figure 4.6 Mean thermal sensation of subjects participating in Test-series-1

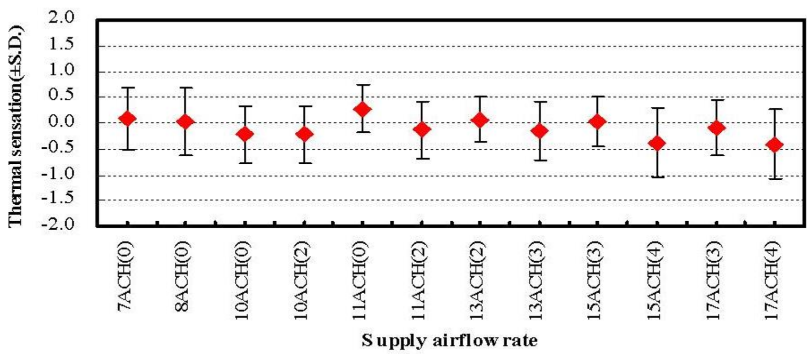


Figure 4.7 Mean thermal sensation of subjects participating in Test-series-2

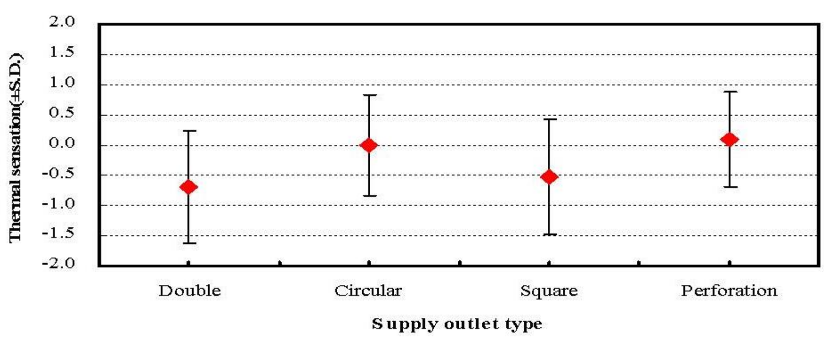


Figure 4.8 Mean thermal sensation of subjects participating in Test-series-3

4.3.5 Applicability of *PMV* model

The most common thermal sensation model using the predicted mean vote (*PMV*) was derived from the steady-state heat balance of a human body (Fanger 1972). This model was based on uniform and steady-state thermal environments. It can predict thermal sensation quite well under those conditions. The detailed formulas of this model are available in BSEN ISO 7730 (2005). As discussion above, stratum ventilation demonstrates the uniform distributions of air temperature and velocity in the occupied zone. The *PMV* model is therefore adopted for stratum ventilation for evaluation. Figure 4.9 shows the comparisons of actual mean thermal sensation votes (ATS) and the predicted values by the *PMV* model for all cases in Test-series-1, 2 and 3. For Test-series-1, the magnitude of the maximum discrepancy between the ATS and *PMV* is only 0.36. This value is 0.44 for Test-series-2. For Test-series-3, the values for the 0.1m and 1.1m levels are 0.83 and 0.62 respectively. The maximum deviations are all smaller than one-scale unit in ASHRAE 7-point thermal sensation scale. This shows that the *PMV* predictions agree reasonably well with the ATS, especially at breathing zone of human subject at 1.1 m AFFL.

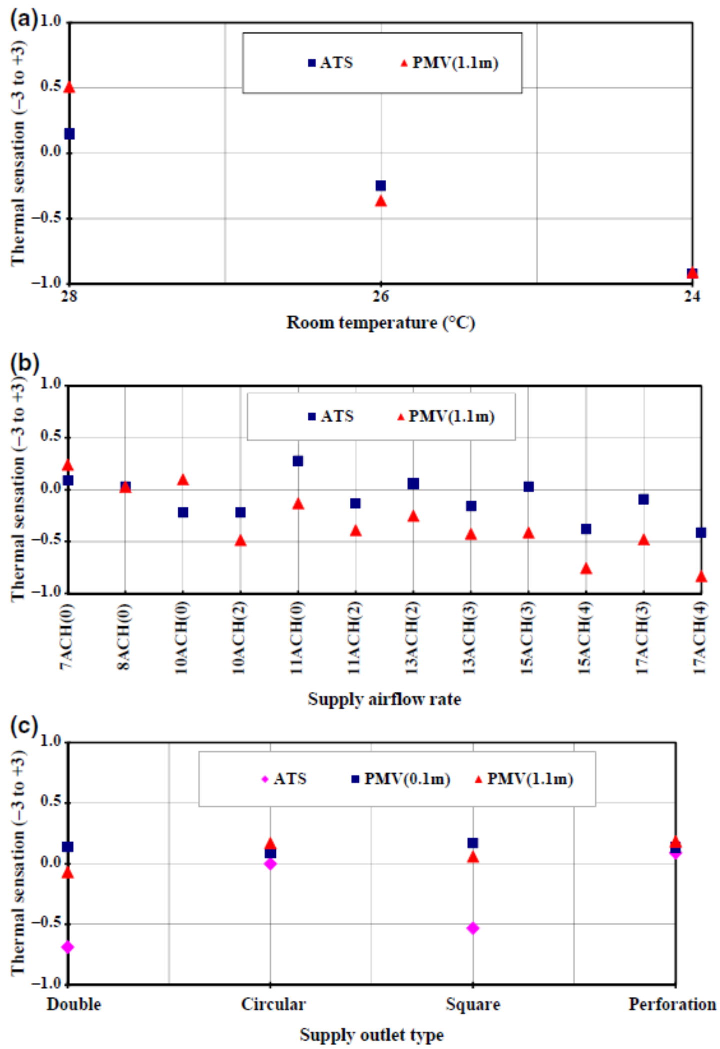


Figure 4.9 Comparison of actual mean thermal sensation (ATS) and predicted mean vote (PMV). (a) Test-series-1, (b) Test-series-2, (c) Test-series-3

4.4 Concluding Remark

In this analysis, data from three separate test series were used to evaluate the uniformity of thermal environments in the occupied zone in a stratum ventilated room by three investigators. For most cases studied, the values of ADPI derived from measured data are not less than 80%. This result shows that thermal comfort condition in the occupied zone is uniform, but excessive supply airflow volume should be avoided.

Thermal sensation on ASHRAE 7-point scale has been also assessed subjectively. The results indicate that standard deviations of thermal sensations of subjects in thermal environments with stratum ventilation are comparable to those with mixing ventilation. This demonstrates thermal sensations of subjects in stratum ventilation are uniformly distributed. This study also experimentally confirmed that stratum ventilation can realise thermal comfort under higher nominal room temperatures; therefore, it is easier to meet the requirements of elevated room temperatures recently imposed by East Asia countries. By comparing ATS with the predicted values of *PMV*, it is found that the deviations are small. The *PMV* model could therefore be used to predict thermal sensation in stratum ventilated rooms.

Both objective experimental measurements and subjective human comfort tests suggest that the thermal environment in the occupied zone of a stratum ventilated room is adequately uniform. The confirmation of the uniform thermal environment served by stratum ventilation can have a number of implications for both quantifying and designing the thermal comfort in a stratum ventilated room. With respect to the former, it provides scientific basis for using existing thermal comfort parameters to describe stratum ventilation; and with respect to the latter, it enables, to large extent, adopting conventional methodology for air distribution design.

CHAPTER 5: COST EFFECTIVENESS ANALYSIS OF STRATUM VENTILATION

In this chapter, a specific goal is to evaluate the economic benefits of adopting the various ventilation systems in the HKSAR typical classroom environment of the warm climate region from stage of cradle to as-built. Life cycle assessment (LCA) is a quantitative method to access the environmental impacts of building systems/products from cradle-to-grave, and requires the use of life cycle costing (LCC) to access the economic and financial viability of investments. LCA and LCC are used to underpin investment decisions in the interest of sustainable development for a real institutional classroom of HKSAR. There are three ventilation systems including mixing, displacement and modified-stratum-3 ventilation, as discussed in the above chapters, being made use of to assess their years of payback in terms of cost investment (CPBP), energy consumption (EPBP), and greenhouse gas emission (GPBP), making reference to the local industry practices. The prediction of system operation performance is based on cooling load estimation and industrial data interpretation, with the scope of energy saving restricted to the life-long electricity saved for the support of system operation, and for reducing the air conditioning system requirements. System boundaries are also carefully defined to facilitate meaningful investigations as below.

5.1 Background of life cycle assessment

Life Cycle Assessment (LCA) aims to evaluate the potential environmental impacts of a product or services during it entire life cycle. According to the ISO 14040 and 14044 standards, a Life Cycle Assessment is carried out in four interdependent distinct phases as shown in Figure 5.1. A complete Life Cycle Assessment study shall comprise the following four main phrases: 1. Goal and scope definition; 2. Life cycle inventory; 3. Life cycle impact assessment; 4. Interpretation of results

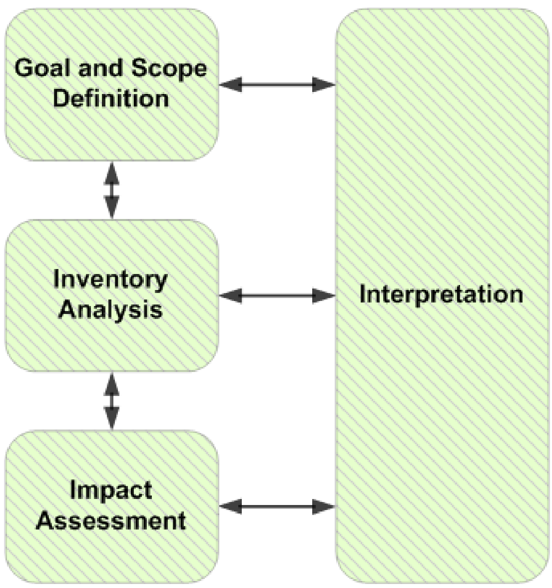


Figure 5.1 Four distinct phases in life cycle assessment

5.1.1 Goal and scope definition

The following definitions are based on the relative ISO 14040 & 14044 standards to be used in this study. "Cradle-to-grave" is the full Life Cycle Assessment from resource extraction "'cradle" to use phase and disposal phase "'grave". All inputs and outputs are considered for all the phases of the life cycle. "Cradle-to-gate" is an assessment of a partial product life cycle from resource extraction “cradle” to the factory gate (i.e., before it is transported to the consumer). The use phase and disposal phase of the product are omitted in this case. One of the significant uses of the cradle-to-gate approach compiles the life cycle inventory (LCI) using cradle-to-gate. This allows the LCA to collect all of the impacts leading up to resources being purchased by the facility. They can then add the steps involved in their transport to plant and manufacture process to more easily produce their own cradle-to-gate values for their products. Cradle-to-as-built is involved the cradle-to-gate stage, installation phase and the related transportation from factory to site installation. It is illustrated in Figure 5.3.

5.1.2 Life cycle inventory

Every key entry of life cycle inventory of energy going in and pollutions emission coming out of the entire process in the preparation of the various ventilation system components are listed. In past years, the specific inventories’ values are referenced from the Life Cycle Assessment tool’s inventory database. Table 5.1 summarizes the inventory database in different countries and regions in respect with defined system boundary, which include the Ecoinvent, ELCD, CLCD, ICE, and LCI.

Table 5.1 Life cycle carbon inventory in different countries and regions

Region	Name of Database	Developer	System Boundary
Switzerland	Ecoinvent	Swiss Centre for Life Cycle Inventories	Gate to Gate
Europe	ELCD (European reference Life Cycle Database)	European Union	Cradle to Gate
Chinese Mainland	"eBalance" (CLCD -Chinese reference Life Cycle Database)	Sichuan University	Cradle to Gate
United Kingdom	ICE (Inventory of Carbon and Energy)	University of Bath, UK	Cradle to Gate
HKSAR	LCI (Life cycle Inventory)	Electrical and Mechanical Service Department, HKSAR Government (EMSD)	Cradle to As-built

The embodied energy values are extracted from the eBalance and EMSD, which are full-featured LCA software developed in China and HKSAR. The "eBalance" LCA software encompasses the Chinese Life Cycle Database (CLCD) which is developed by Sichuan University and IKE Environmental Technology. The database is tailor-made to represent the Chinese market average technology. It also records the Ecoinvent database and the European Reference Life Cycle Database (ELCD). Since the free version of the "eBalance" software only provides limited data from (CLCD), some of the embodied energy data is referenced from the Ecoinvent and (ELCD) database stored in the "eBalance" software. The Ecoinvent and (ELCD) database support the inventory data of materials that are imported to China and provide referencing inventory values that are not yet available in the China database. LCI data developed by EMSD bases evaluate the direct and indirect impact of the material manufacturing process using the life cycle assessment technique. Inventory of this study is based on the surveyed LCA/LCC data which were conducted on components and materials used in 28 completed commercial buildings in HKSAR under the life cycle energy analysis (LCEA) study software, retrieved from the EMSD website. It can identify the range of components and materials that would dominate the total environmental impacts of buildings. Other well-known Life Cycle Assessment programs for buildings include: ATHENA (Canada); BEES (US); and ENVEST (UK), etc.

5.1.3 Life cycle impact assessment

Inventory data is summed up together into aggregate indicators, such as cumulative greenhouse gas production and energy consumption. These data can conduct additional impacts that will be incurred during the construction stage; and the countries from which building materials and services equipment are imported into HKSAR for building construction. Other supporting data has also been available, such as the type of fuel for production processes, the fuel-mix used in different countries for electricity generation and the distance of transport involved for importing system and equipment components and materials from concern countries, etc.

In the calculations of specific indicators, weighting factors are needed to compute the equivalent pollution emission or energy resource. Regarding the current case, the weighing factors for methane (CH_4) and nitrous oxide (N_2O) are 25 and 298, since 1 kg of methane and 1 kg of nitrous oxide contribute the same greenhouse effect as 25 kg and 298 kg of carbon dioxide respectively. Internationally, the idea of LCA has been raised in: Earth Summit at Rio; Kyoto Protocol; UNEP; Earth Summit at Johannesburg. These data using in special tools are designed for assessing different types of buildings, and emphasizing on different phases of the life cycle were studied by Haapio and Viitaniemi (2008) in order to address an environmental aspects, sustainable building includes economic and social aspects.

In Figure 5.2, Life cycle impact assessment follows the LCI analysis can subdivide into 5 steps: (1) first categorizes the impacts on resources consumption and emissions; (2) Characterize the equivalent amount of CO_2 under the global warming category by converting the quantities of various types of impacts under each category into equivalent quantities of a reference impact, e.g. methane; (3) Normalize the impact profile into a set of dimensionless numbers. A normalized impact indicator reflects the proportional contribution of the product to the total impact of the same type in the region, and hence the seriousness of the impact the product would incur; (4) Group the convenience of further study proceeding and data analysis; (5) weight the normalized impact indicators.

In this study the weighting factors will not apply. In addition to apply impact indicator of life cycle energy use including embodied and operating energy use will be studied. The normalization factors used are based on 1 TJ of electricity generation in HKSAR. This study is focus on the comparison of stratum and displacement ventilation, with reference to mixing ventilation during supply and operating path.

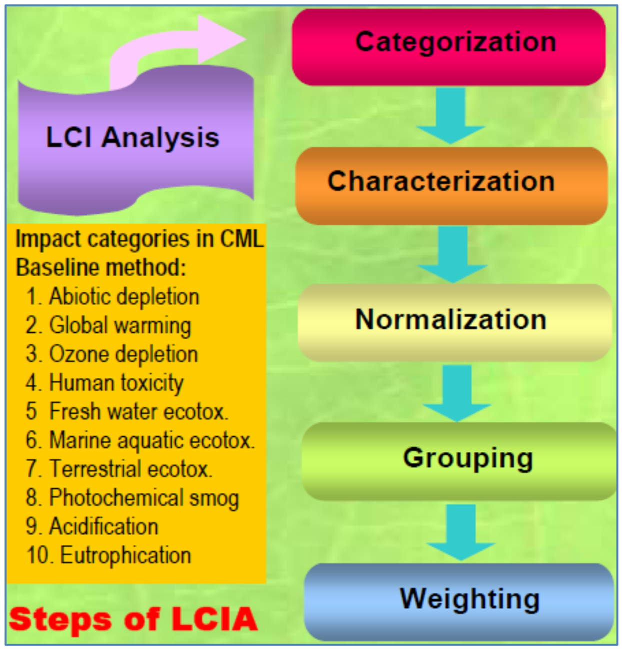


Figure 5.2 Steps of life cycle inventory analysis

(extracted from the user manual under the EMSD website)

5.1.4 Interpretation of results

The final phase summarizes the key Figures and compares the results with the initial goal. The influences to the final result due to limitations (such as the unavailability of the specific data inventories) can be addressed and the degree of influence to the accuracy of the final results is analyzed and discussed in the following section.

5.1.5 System boundary of the fife cycle assessment study

Figure 5.3 presents the system boundary of the LCA study for the incorporation of Mixing, Displacement, and Stratum ventilation in a real case. To simplify the full boundary, disposal process for the system product is excluded in cradle-to-grave assessment. A full boundary shall include the processes from raw material extraction to the system fabrication, and to the final disposal stage at the end of system service life.

The LCA process is a traditional way of analyzing building system life cycle emissions. The principle is to calculate GHGs of each process of the building system life cycle individually in order to form a chain of the processes that covers the whole life cycle (Bilec, 2007). Each process analysis is conducted using process-specific primary (i.e., material and energy flows in the manufacturing process) and secondary data (i.e., amount of GHG emissions per manufacturing process), which lead to very accurate results of the modelling. However, there is nearly an indefinite number of single processes in a building system life cycle, and including all of them in the modelling is practically impossible. This problematic characteristic of process LCA modelling is known as a truncation problem (Suh et al. 2004). A process LCA practitioner has to define a border that separates the processes included in the modelling from those that are left out of it. Thus, it is probable that significant processes are also left out of the modelling along with the insignificant ones. Process LCAs are also very laborious and require a large amount of data since secondary data has to be acquired separately for each process.

Using cradle-to-gate approach compiled the life cycle inventory; this allows the LCA to collect all of the impacts leading up to resources being purchased. The raw materials for the fabrication of the various ventilation system components as well can be mainly mined in China with the rest imported from other places. For the real application case in HKSAR, all the main parts and sub-parts of the ventilation systems are considered to be fabricated in mainland China. So the transportation of raw materials within China or from worldwide to the by or main-product fabrication factories in China are taking place; All the components of the ventilation system are fabricated and assembled in China and are delivered from the main factory in China to the installation site in HKSAR. The ventilation system is finally installed on the classroom in HKSAR. Each component of system, the electrical wiring works and the water piping works are considered in this stage. The disposal of the ventilation system is excluded from the life cycle assessment. If such a process is considered, the energy demand and greenhouse gas emission for reuse, recycling, landfill and incineration have to be added in the total inventories summation. In some cases, this part is ignored due to the lack of information on the disposal treatment and their inventories data. This process is also omitted in the current study.

5.2 Payback period of embodied energy and greenhouse gas emission

Based on the system boundary as specified, the energy payback period (EPBP) of the different ventilation system can be calculated by the following equation:

$$\mathrm{EPBT} = \frac{\sum AHU + \sum Installation + \sum Transportation}{E_{AirSide} + E_{WaterSider} - E_{OM}} \qquad (5.1)$$

where $\sum AHU$ is the embodied energy of the air handling unit from 'Cradle-to-Gate' which means that the production processes from extraction of raw materials, to fabrication and leaving the production factory ultimately; $\sum Installation$ is the embodied energy of all accessories of different ventilation system erected on site. $\sum Transportation$ is the embodied energy of transportation of the different ventilation system from the factory to installation site. In the cases of mixing, displacement and stratum ventilation, the factory location is chosen in Dongguang, where is a main industrial region in southern China. The final installation site is in HKSAR; $E_{AirSide}$ *and* $E_{WaterSide}$ are the annual energy saving in air side equipment and water equipment respectively. E_{OM} is the annual operation and maintenance energy difference for displacement ventilation and mixing ventilation in comparison with mixing ventilation as reference case. The amount of energy for operation and maintenance is assumed as 6% of the energy use in supply and installation phase.

Consequently, with equation (5.1), equation (5.2) can be re-written as:

$$\mathrm{EPBP} = \frac{\sum AHU + \sum installation + \sum Transportation}{E_{AirSide} + E_{WaterSider} - (\sum AHU + \sum installation + \sum Transportation) x 0.06} \tag{5.2}$$

Similarly, the greenhouse gas emission payback period can be expressed as

$$\mathrm{GPBP} = \frac{\Omega_{AHU} + \Omega_{Installation} + \Omega_{Transportation}}{Z_{AirSide} + Z_{WaterSide} - (\Omega_{AHU} + \Omega_{Installation} + \Omega_{Transportation}) x 0.06} \tag{5.3}$$

where Ω_{AHU}, $\Omega_{Installation}$ and $\Omega_{Transporation}$ are respectively the amount of greenhouse gas

(GHG) emission during the complete manufacturing process of the air handling unit, the GHG emission for all necessary accessories of different ventilation installed on site, and the GHG emission due to the transportation of the system components from the ultimate factory to the installation site respectively; $Z_{AirSide}$ and $Z_{WaterSide}$ in equation (5.3) are the annual reduction of GHG emission from the local utility service due to the saving of energy use by adopting the alterative ventilation design. Again, the term due to operation and maintenance is assumed 6% of figure in supply and installation phase.

5.3. Research process

5.3.1. Boundary condition of MV, DV & SV

This cost effectiveness analysis of stratum is based on the results of thermal neutral temperature from forty-eight subjects participated in all ASHRAE 7-point scale tests under the same boundary conditions for mixing, displacement and stratum ventilation methods in the environmental chamber. 48 male and female subjects under the combination of 10 & 15 ACH and 24℃, 26℃ & 28℃ are used to determine the neutral temperature by regression analysis methodology. The neutral temperatures of young HKSAR Chinese under mixing ventilation, displacement ventilation, and stratum ventilation are found to be 24.6℃, 25.1℃, and 27.1℃ at 10 ACH are used for this cost effectiveness study. The result indicated that stratum ventilation could provide satisfactory thermal comfort level to rooms of temperature up to 27℃.

5.3.2. Defining energy parameter

The energy options to be evaluated are the base case scenario along with alternative options, where different ventilation systems are combined under their estimated neutral temperature. Fong et al. (2011) estimated the neutral temperature of mixing, displacement and stratum are 24.6℃, 25.1℃ and 27.1℃ at 10 ACH respectively. The design parameters, such as *COP* of chiller, internal heat sources are mentioned in Table 3.7 & 3.9 respectively in the previous chapter 3.5, are used to study a year-round sensible energy consumption of mixing, displacement and stratum ventilation. Total cooling sensible energy saving at 10 ACH and 15 ACH are in the range of 4.08% to 18.84% and 3.99% to 19.97% in comparison with mixing ventilation as shown in Table 3.10b. The saving percentage of displacement and stratum ventilation of 4.08% and 18.84%, in comparison with mixing ventilation at 10ACH, will be used in LCA & LCC afterward.

5.4 Analysis and discussion

5.4.1 Acquiring case data

Analysis employed all the acquired cost data (Table 5.2 to 5.11) during the life cycle for three ventilation systems. By applying the economic criteria, the current net cost could be attained. The LCA analysis of the study utilizes supply and installation costs of ventilation systems plus energy consumption amounts as initial data for the assessment of carbon emissions. Costs of maintenance, repair and replacement plus amounts of energy consumed in the operation phase of the ventilation systems were used as initial data in the use-phase assessment. In order to make a comprehensive assessment, different time perspectives were used in the Life Cycle Assessment. The chosen time periods were 5, 10, 15 and 20 years.

The life cycle costing is divided into (1) Supply and installation and (2) use-phase assessment. In the construction part, the initial investment costs are attained. In the use-phase assessment, energy consumption, operational costs, maintenance costs and building-related replacement schedules are estimated. Additionally, relevant economic criteria for LCC are defined.

5.4.1.1 Supply and Installation phase assessment.

The first step is to acquire relevant cost data for the evaluation of the different ventilation methods. The focus of the LCC analysis was to compare different components of each ventilation system. The cost data is attained from the cost estimators of the contractor and supplier as tabulated in Table 5.2 to Table 5.4. The different arrangement of ductwork, diffuser, size and layout for mixing, displacement and stratum ventilation are illustrated in Figure 5.4, 5.5 and 5.6 respectively. It will form as basic for this cost, embodied energy and GHG emission analysis.

5.4.1.2 Use phase assessment

The next step was to estimate the energy demand of each ventilation system. Maintenance costs for the different ventilation systems were assumed as same percentage, 6% of initial cost for this study. Together, these conclude the annual maintenance costs. An estimated salvage cost for the life cycle works was the last piece of data needed for the assessment as well.

Return Air duct	Horizontal		Vertical	
300X300 RAD	300*1800*2	1.08	300*1800*2	1.08
300X300 RAD	300*900*2	0.54	300*900*2	0.54
300X200 RAD	300*900*2	0.54	200*900*2	0.36
Damper	500*480*2	0.48	500*480*2	0.48
400x300 RAD	400*2600*2	2.08	300*2600*2	1.56
	Sum	4.72		4.02
	Total Area (m^2)	8.74		
Supply Air Duct	Horizontal		Vertical	
400x250 SAD	400*3500*2	2.8	250*3500*2	1.75
300X250 SAD	300*2650*2	1.59	250*2650*2	1.325
Middle SAD	400*3500*2	2.8	250*3500*2	1.75
300X250 SAD	300*2650*2	1.59	250*2650*2	1.325
	Sum	8.78		6.15
	Total ductwork Area (m^2)	14.93		

	RETURN AIR GRILLE		SUPPLY AIR TERMINAL	
	600x600		600x600	
Dimension	0.6	0.6	0.6	0.6
Number	3		6	
Total diffuser area (m^2):	3.24			

Ventilation Modes	Level of ventilation outlet		Mode of dampers operation (Location of each damper shall refer to SK-S-01)												
	Supply air	Return air	MD-01-S	MD-02-S	MD-03-S	MD-04-S	MD-05-S	MD-06-S	MD-07-S	MD-08-S	MD-01-R	MD-02-R	MD-03-R	MD-01	MD-02
1. Well mixing	Ceiling	Ceiling	Open	Open	Close	Close	Close	Close	Close	Close	Open	Close	Close	Close	Close

Figure 5.4 (a) Ductwork and diffuser size for Mixing Ventilation

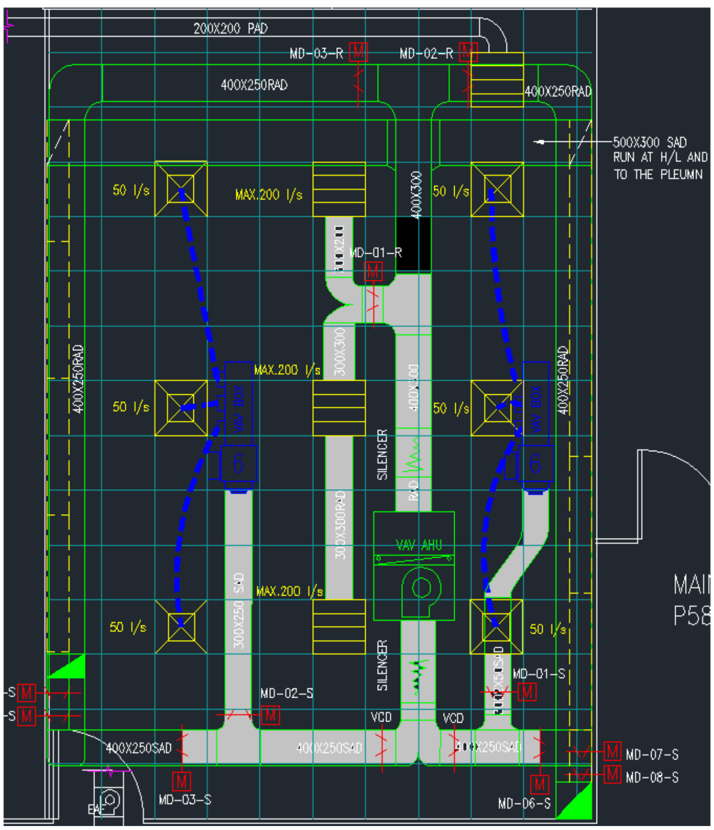


Figure 5.4 (b) Ductwork and diffuser layout plan of Mixing Ventilation

	Horizontal					Vertical			T1	T2
Dimension	500x250	400x250	300x300	300x200	400x300	500x250	400x300			
Ductwork Length	1103	3538	3635	1268	1933	1150	1150			
		7089			244					
					640					
		1724								
		7663								
Sum of length	1103	20014	3635	1268	2817	1150	1150	0		
Perimeter	1.5	1.3	1.2	1	1.4	1.5	1.4			
Area	1.6545	26.0182	4.362	1.268	3.9438	1.725	1.61	0	0.66875	0.6

Total ductwork area (m^2)	41.85025
Total diffuser area (m^2)	1.8

	RETURN AIR GRILLE		SUPPLY AIR TERMINAL	
Dimension	600x600		300x300	
	0.6	0.6	0.3	0.3
Number	3		8	

Figure 5.5 (a) Ductwork and diffuser use for Displacement Ventilation

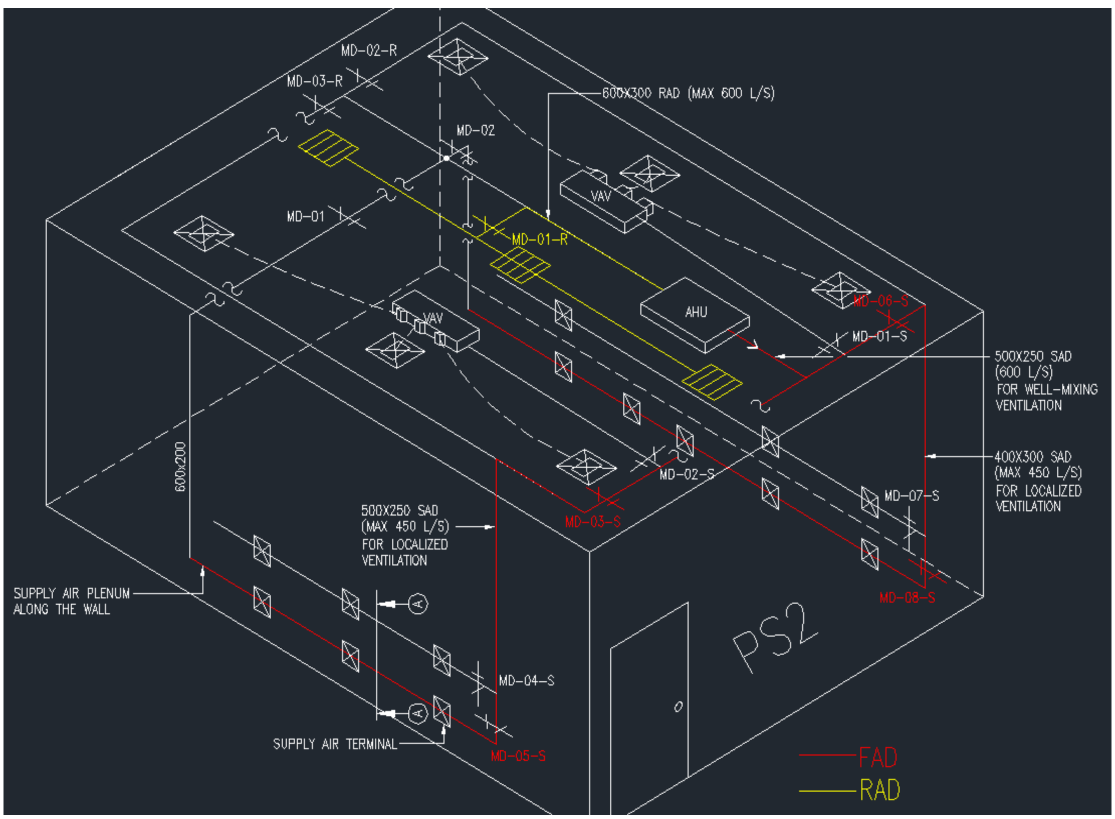


Figure 5.5 (b) Overview of ductwork and diffuser uses for Displacement Ventilation

	Stright pipe									
	Supply Air					Return Air				
	From AHU to MD-06	From MD-06 to MD-07				From MD-07 to MD-04	From MD-04 to MD-03			From MD-03 to AHU
Dimension (mm x mm)	400x250	400x250	400x250	410x400	300x300	300x300	400x300	400x250	400x250	400x300
Ductwork Length (m)	1.103	1.314	0.972	1.15	6.075	4.65	1.15	6.727	3.739	4.9
Perimeter (m)	1.3	1.3	1.3	1.62	1.2	1.2	1.4	1.3	1.3	1.4
Area	1.4339	1.7082	1.2636	1.863	7.29	5.58	1.61	8.7451	4.8607	6.86

	Elbow piece										
	Supply Air								Return Air		
	From AHU to MD-06						From MD-06 to MD-07		From MD-03 to AHU		
Dimension (mm x mm)	250x100	250x51	250x250	250x150	500x500	400x250	400x250	400x250	810x400	810x250	400x250
Number of piece	2	2	2	2	2	1	2	2	1	1	1
Perimeter (m)	0.025	0.01275	0.0625	0.0375	0.25	0.1	0.1	0.1	0.324	0.2025	0.1
Area	0.05	0.0255	0.125	0.075	0.5	0.1	0.2	0.2	0.324	0.2025	0.1

	RETURN AIR GRILLE		SUPPLY AIR TERMINAL	
Dimension	300x300		300x300	
	0.3	0.3	0.3	0.3
Number	4		4	

Total ductwork area(m^2)	43.12
Total diffuser area(m^2)	0.72

Figure 5.6 (a) Ductwork and diffuser use for Stratum Ventilation

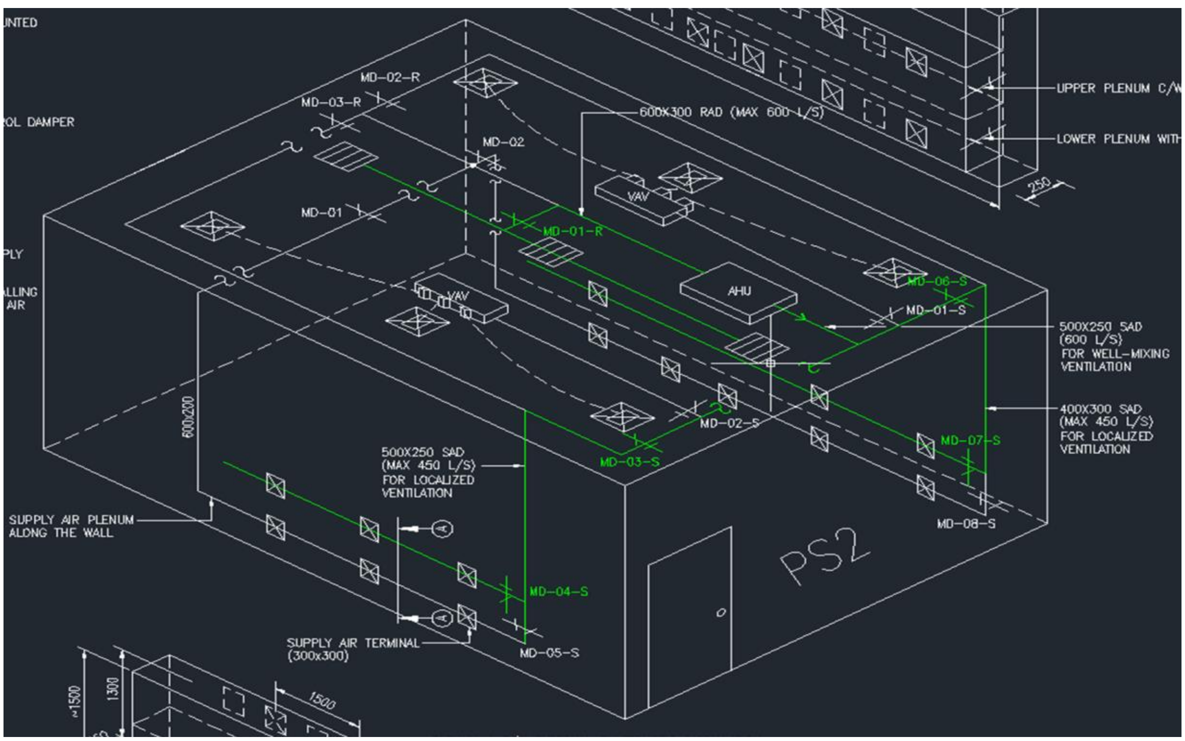


Figure 5.6 (b) Overall view of ductwork and diffuser installation of Stratum Ventilation

5.4.1.3 Relevant economic parameters

The results of initial cost analysis as shown in item A1, A2, A3 of the following Table 5.2, 5.3 and 5.4 respectively, presented show that there is higher initial investment in alternatives displacement (5.91% more) and stratum (5.71% more) ventilation methods.

When all the relevant cost data for the assessment was received, it had to be placed into the life cycle framework (service years). In order to make the different ventilation methods comparable with each other, the cost data had to be indexed and discounted so that each ventilation method could be presented in a net present value (NPV) context and be comparable with each other from a life cycle perspective as shown in Table 5.6 to Table 5.8.

Cost-effectiveness compared their NPV of the annual costs and benefits over the service life of three ventilation systems are summarized in Table 5.9.

Table 5.2 Cost Analysis of Mixing Ventilation

A	Initial Cost	Quantity	Unit	HK$ /Unit	HK$
1	Supply and install air handling unit completed with double skin 50 mm thickness panels, 50mm polyurethane insulation, 50mm aluminum filter, 450 L/s supply air fan.	1	no.	50,000	50,000
2	Supply and install galvanized iron air ductwork with accessories (*refer to Figure 5.5 for detail)	*14.93**	m2	300	4,479
3	Supply and install 25mm thickness, 40 kg/m3phenolic foam thermal insulation for air ductwork	*14.93**	m2	180	2,687
4	Supply and install aluminum supply air grille (SAG) completed with volume control damper 200x200mm wall mounted SAG, 600x600mm ceiling mounted SAG	6	no.	450	2,700
5	Supply and install aluminum return air louver (RAL): 200x200mm wall mounted RAL, 600x600mm ceiling mounted RAL	3	no.	450	1,350
6	Supply and install a framework support for ceiling mounted AHU	1	lot	6,000	6,000
7	Supply and install duct silencer	2	no.	4,000	8,000
8	Supply and install 50mm diameter mild steel chilled water pipework completed with phenolic foam thermal insulation and connection to air handling unit	30	m	450	13,500
9	Supply and install 32mm diameter galvanized steel condensate drain pipework completed with phenolic foam thermal insulation	20	m	320	6,400
10	Supply and install electrical power supply and control including starter panel, switch and wiring	1	lot	50,000	50,000
11	Supply and install Direct Digital Controller (DDC) control equipment including control valve, temperature sensor, relative humidity sensor, pressure sensor, flow sensor, actuator for monitoring and control of the air handling unit	1	lot	70,000	70,000
	Sub- total A (HK$):			A1:	215,116 (base)
B	Annual Operating Cost			B1:	
	Year-round energy consumptions at 10 air change per hour and the design parameters in Table 3.11.				64,737
C	Annual Maintenance Cost (assume 6% of initial cost)			C1:	12,907
		A1+B1+C1 =			292,760

Table 5.3 Cost Analysis of Displacement Ventilation

A	Initial Cost	Quantity	Unit	HK$ /Unit	HK$
1	Supply and install air handling unit completed with double skin 50 mm thickness panels, 50mm polyurethane insulation, 50mm aluminum filter, 450 L/s supply air fan.	1	no.	50,000	50,000
2	Supply and install galvanized iron air ductwork with accessories (*refer to Figure 5.6 for detail)	*41.85**	m2	300	12,555
3	Supply and install 25mm thickness, 40 kg/m3phenolic foam thermal insulation for air ductwork	*41.85**	m2	180	7,533
4	Supply and install aluminum supply air grille (SAG) completed with volume control damper 200x200mm wall mounted SAG 600x600mm ceiling mounted SAG	8	no.	350	2,800
5	Supply and install aluminum return air louver (RAL) 200x200mm wall mounted RAL 600x600mm ceiling mounted RAL	3	no.	350	1,050
6	Supply and install a framework support for ceiling mounted AHU	1	lot	6,000	6,000
7	Supply and install duct silencer	2	no.	4,000	8,000
8	Supply and install 50mm diameter mild steel chilled water pipework completed with phenolic foam thermal insulation and connection to air handling unit	30	m	450	13,500
9	Supply and install 32mm diameter galvanized steel condensate drain pipework completed with phenolic foam thermal insulation	20	m	320	6,400
10	Supply and install electrical power supply and control including starter panel, switch and wiring	1	lot	50,000	50,000
11	Supply and install Direct Digital Controller (DDC) control equipment including control valve, temperature sensor, relative humidity sensor, pressure sensor, flow sensor, actuator for monitoring and control of the air handling unit	1	lot	70,000	70,000
	Sub- total A (HK$):			A2:	227,838 (+5.91%)
B	Annual Operating Cost			B2:	
	Year-round energy consumptions at 10 air change per hour and the design parameters in Table 3.11				62,198 (-4.08%)
C	Annual Maintenance Cost (assume 6% of initial cost)			C2:	13,670
		A2+B2+C2 =			303,706 (3.74%)

Table 5.4 Cost Analysis of Stratum Ventilation

A	Initial Cost	Quantity	Unit	HK$ /Unit	HK$
1	Supply and install air handling unit completed with double skin 50 mm thickness panels, 50mm polyurethane insulation, 50mm aluminum filter, 450 L/s supply air fan.	1	no.	50,000	50,000
2	Supply and install galvanized iron air ductwork with accessories (*refer to Figure 5.7 for detail)	*43.12**	m2	300	12,936
3	Supply and install 25mm thickness, 40 kg/m3phenolic foam thermal insulation for air ductwork	*43.12**	m2	180	7,762
4	Supply and install aluminum supply air grille (SAG) completed with volume control damper: 200x200mm wall mounted SAG 600x600mm ceiling mounted SAG	4	no.	350	1,400
5	Supply and install aluminum return air louver (RAL): 200x200mm wall mounted RAL 600x600mm ceiling mounted RAL	4	no.	350	1,400
6	Supply and install a framework support for ceiling mounted AHU	1	lot	6,000	6,000
7	Supply and install duct silencer	2	no.	4,000	8,000
8	Supply and install 50mm diameter mild steel chilled water pipework completed with phenolic foam thermal insulation and connection to air handling unit	30	m	450	13,500
9	Supply and install 32mm diameter galvanized steel condensate drain pipework completed with phenolic foam thermal insulation	20	m	320	6,400
10	Supply and install electrical power supply and control including starter panel, switch and wiring	1	lot	50,000	50,000
11	Supply and install Direct Digital Controller (DDC) control equipment including control valve, temperature sensor, relative humidity sensor, pressure sensor, flow sensor, actuator for monitoring and control of the air handling unit	1	lot	70,000	70,000
	Sub- total A (HK$):			A3:	227,398 (+5.71%)
B	Annual Operating Cost			B3:	
	Year-round energy consumptions at 10 air change per hour and the design parameters in Table 3.11.				54,475 (-18.84%)
C	Annual Maintenance Cost (assume 6% of initial cost)			C3:	13,644
		A3+B3+C3 =			295,519 (0.94%)

The discount rate of 15% was tied to an average annual energy cost of fuel oil, liquefied petroleum gas (LPG) and natural gas in between 2002 to 2012 as shown in Table 5.5. It is extracted from HKSAR Energy Statistics 2012 Annual Report, HKSAR. In Figure 5.7, the indexation rate of 4.3% was predicted by the average annual charge on consumer price index in last two years as shown in Table 5.6. These rates are applied to construction costs, energy costs, maintenance costs and life cycle costs. In the analysis, the energy indexation will be in focus on peak sensible space load, excluding any latent load in order to simplify the comparison, when analyzing the different ventilation system options. The Life Cycle Cost (LCC) assumes its life of 5, 10, 15 & 20 years and the salvage values of the system will be 50%, 25%, 10% and 0% respectively of their initial cost. Thus, the system life time is assumed as 20 years for these ventilation comparisons study.

Table 5.5 Unit values of imports of oil

Year	Fuel oil	Percentage change (%)	LPG (HK$/Kg)	Percentage change (%)	Natural Gas (HK$/Kg)	Percentage change (%)
2002	1.17	base	2.27	base	1.09	base
2003	1.35	15.38	2.64	16.30	1.13	3.67
2004	1.44	6.67	3.11	17.80	1.01	-10.62
2005	2.02	40.28	3.83	23.15	1.05	3.96
2006	2.45	21.29	4.47	16.71	1.16	10.48
2007	2.78	13.47	5.19	16.11	1.54	32.76
2008	4.11	47.84	6.91	33.14	1.78	15.58
2009	2.71	-34.06	4.49	-35.02	1.77	-0.56
2010	3.58	32.10	5.95	32.52	2.33	31.64
2011	4.94	37.99	7.42	24.71	2.57	10.30
2012	5.21	5.47	7.84	5.66	3.13	21.79
Average annual energy cost (% change) in between 2002 to 2012						
		18.64		15.11		11.90

Figure 5.7: HKSAR inflation rate forecast (Source from HKSAR inflation: http://www.tradingeconomics.com/hong-kong/inflation-cpi)

5.4.1.4 Net present value for life cycle assessment

Net Present value (NPV) of life cycle assessment of three ventilation systems in 5, 10, 15 & 20 service year has been taken into account of discounting and inflation factor. An expense of one dollar at the end of the first time period, when inflated at rate of k per time period, will be $(1+k)^{n-1}$ at the end of the nth time period. The present value (PV) of this equivalent one dollar at the end of the nth period is $\frac{(1+k)^{n-1}}{(1+i)^j}$. Present value (PV) factor of the series of "n" such payment applying for inflation recurred in each year can be found by summing the expression over n periods, i.e. PV factor $(n,k,i) = \sum_{j=1}^{n} \frac{(1+k)^{j-1}}{(1+i)^j}$; if $i \neq k$, $\text{PV factor}(n,k,i) = \frac{1}{i-k}\left(1 - \frac{(1+k)^n}{(1+i)^n}\right)$; if $i = k$, $\text{PV factor} = \frac{n}{1+k}$.

Table 5.6 Life cycle NPV costing analysis of mixing ventilation during 20 service years

Net present value (NPV)	Mixing Ventilation			
[1] Initial Cost, IC	*215,116*			
Annual Energy Cost, EC[see Note 1]	64,737			
Annual Maintenance Cost, MC[see Note 2]	12,907			
Life Time (Yr5, Yr10, Yr15, Yr20)	5	10	15	20
Present value (PV) Factor [see Note 3]	4.4995	6.9363	8.1479	8.7502
[2] PV of O&M Cost, = (EC+MC) x PV FACTOR	349,358	538,563	632,632	679,401
Equivalent present worth of F, P_f [see Note 4]	0.4972	0.2472	0.1229	0.0611
" a" % of initial Cost = salvage cost	50%	25%	10%	0%
Salvage value, SC = IC x a%	107,558	53,779	21,512	0.00
[3] PV of Salvage Cost [see Note 2] =SC x P_f	*53,475*	*13,293*	*2,644*	*0.00*
Life Cycle Costing [1] + [2] -[3], HK$	510,999	740,386	845,105	894,517

Table 5.7 Life cycle NPV costing analysis of displacement ventilation during 20 service years

Net present value (NPV)	Displacement Ventilation			
[1] Initial Cost, IC	227,838 (5.91% more than mixing ventilation)			
Annual Energy Cost, EC[see Note 1]	62,198			
Annual Maintenance Cost, MC[see Note 2]	13,670			
Life Time (Yr5, Yr10, Yr15, Yr20)	5	10	15	20
PV factor [see Note 3]	4.4995	6.9363	8.1479	8.7502
[2] Present value of O&M Cost, =(EC+MC) x PV FACTOR	341,368	526,247	618,165	663,864
P_f [see note 4]	0.4972	0.2472	0.1229	0.0611
" a" % of initial Cost = salvage cost	50%	25%	10%	0%
Salvage value, SC = IC x a%	113,919	56,960	22,784	0.00
[3] PV of Salvage Cost,= SC x Pf	56,638	14,080	2,800	0.00
Life Cycle Costing [1] + [2] - [3], HK$	512,568	740,006	843,203	891,702
Percentage difference in comparing with Mixing Ventilation	0.31%	-0.05%	-0.23%	-0.31%

Table 5.8 LCC analysis in NPV (HK$) of stratum ventilation during 20 service years

Net present value (NPV)	Stratum Ventilation			
[1] Initial Cost, IC	227,398 (5.71% more than mixing ventilation)			
Annual Energy Cost, EC[see Note 1]	31,598			
Annual Maintenance Cost, MC[see Note 2]	54,475			
Life Time (Yr5, Yr10, Yr15, Yr20)	5	10	15	20
PV factor [see Note 3]	4.4995	6.9363	8.1479	8.7502
[2] Present value of O&M Cost, =(EC+MC) x PV Factor	306,502	472,497	555,027	596,058
P_f [see Note 4]	0.4972	0.2472	0.1229	0.0611
" a" % of initial Cost = salvage cost	50%	25%	10%	0%
Salvage value, SC = IC x a%	113,699	56,849	22,740	0.00
[3] PV of Salvage Cost,= SC x Pf	56,528	14,052	2,795	0.00
Life Cycle Costing [1] + [2] -[3], HK$	477,371	685,843	779,630	823,456
Percentage difference in comparing with Mixing Ventilation	-6.58%	-7.37%	-7.75%	-7.94%

Note 1 to 4 for the above Table 5.6, 5.7 & 5.8:

Note 1: Assumption of annual energy cost = HK$ 2,628 per kW; = 12 hours/day x 0.6 seasonal factor x 365 days x HK$1/kWh
Note 2: Annual maintenance cost = 6% x Initial cost

Note 3: By substituting annual discount rate, i = 0.15 (i.e.15%) and inflation rate, k = 0.043 (4.3%) into the $(n,k,i) = \sum_{j=1}^{n} \frac{(1+k)^{j-1}}{(1+i)^{j}}$; if $i \neq k$, PV factor$(n,k,i) = \frac{1}{i-k}\left(1 - \frac{(1+k)^{n}}{(1+i)^{n}}\right)$; if $i = k$, PV factor $= \frac{n}{1+k}$ to find out the equivalent present value (PV) factor occurring in "n" years in the future. It is shown in the following Table 5.6, 5.7 & 5.8

Note 4: By substituting annual discount rate, i = 0.15 (i.e.15%) to find out the equivalent present value of F into $P_f = \frac{F}{(1-i)^{n}}$, cash flow occurring in " n" years in the future

Table 5.9 Summary of LCC analysis in NPV (HK$) for service year at 5, 10, 15 & 20

	Net present value (HK$) for Yr5, Yr10, Yr15 & Yr20			
System Life Time (Year)	5	10	15	20
Mixing Ventilation (Reference Case)	510,999	740,386	845,105	894,517
Displacement Ventilation	512,568 0.31%	740,006 -0.05%	843,203 -0.23%	891,702 -0.31%
Stratum Ventilation	477,371 -6.58%	685,843 -7.37%	779,630 -7.75%	823,456 -7.94%

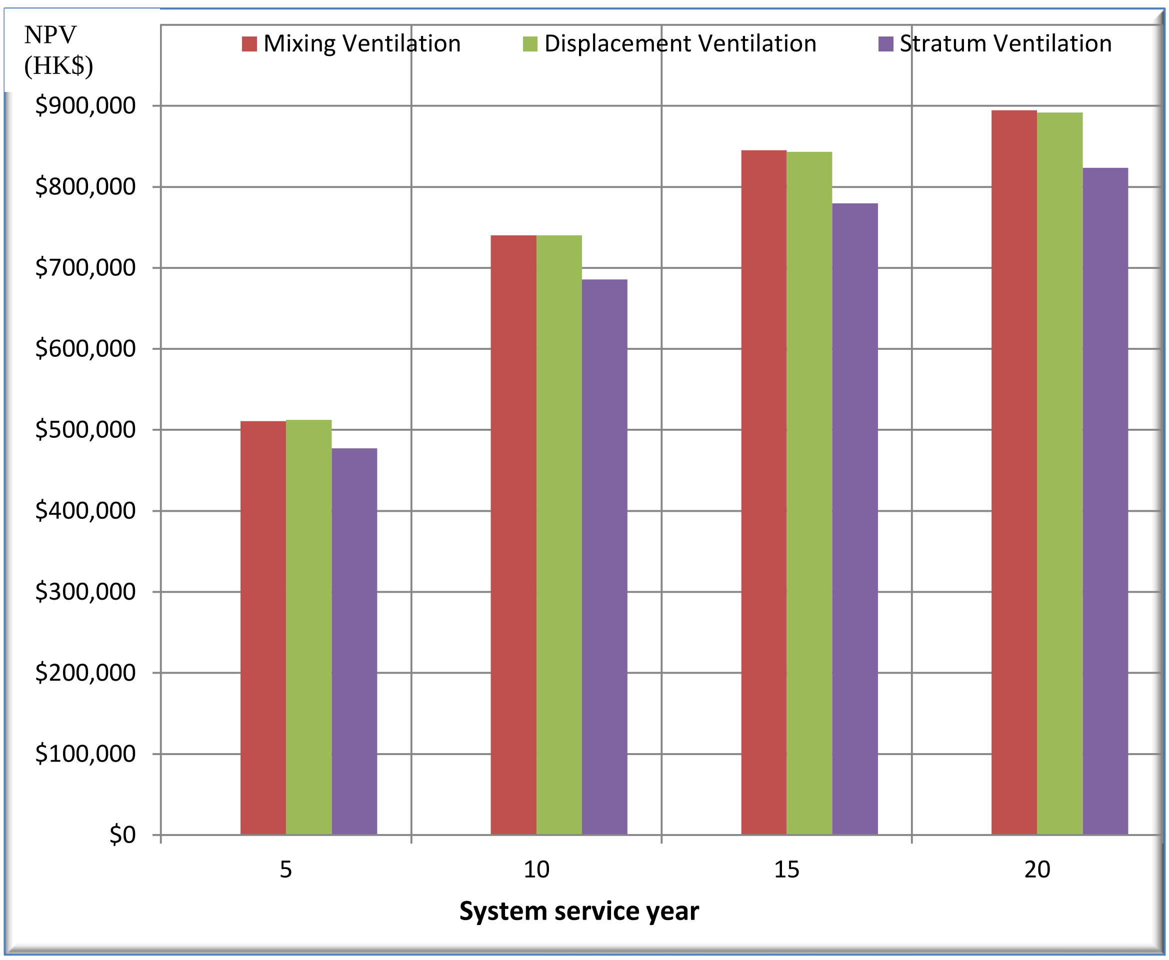


Figure 5.8 Comparison of Net Present value (HK$) of mixing, displacement and stratum ventilation systems in life cycle assessment

As shown on Table 5.9 and Figure 5.8, the descending order of life cycle costing of three ventilation systems is displacement, stratum and mixing. i.e. displacement ventilation is the most expensive option in all of 5,10, 15 & 20 service years. The differential percentage of LCC is found to decline at the end of life time as shown in Table 5.9. The percentages of cost reduction of stratum ventilation in comparing with mixing ventilation are 6.58%, 7.37%, 7.75%, and 7.942% in 5, 10, 15 and 20 service years. This implies that a payback period of stratum ventilation have been feasibly achieved by extending its life span through better maintenance work.

5.4.2 Life cycle cost assessment

In the study, the GHG emissions of different ventilation system life cycle are divided into supply phase, installation phase and use-phase emissions. Recycling process of the end-of-life disposal step (cradle to cradle variant analysis and disposal process for this study) is excluded in cradle-to-grave assessment. It is because of the general system life time are expected to be more than 20 years. The GHG emissions of the supply and installation phase consist of the emissions embodied in the utilized materials as well as those caused by the different ventilation systems. The use-phase emissions consist of GHGs caused by primary energy consumption of the buildings with different ventilation systems as well as GHGs caused by the building maintenance. The supply and installation, and use-phase activities are presented in more detail in the following sections of the study.

5.4.2.1 Supply and installation phase assessment

LCA of the supply and installation phase was carried out using the costs of the construction as primary data apart from electricity and heat, which were assessed using the energy consumption amounts of the construction project. The amounts of energy consumed in the construction process were calculated using the costs of energy and prices of electricity and heat for the construction company. All the primary data of the supply and installation phase was received from the Econivent database.

The GHG emissions caused by supply and installation of various ventilation systems were assessed using the same model of air handling unit. Its mass weight of each component inside air handling unit is shown in Table 5.10. In comparison with different data inventory for GHG emission estimation, it has a derivation between LCI and Ecovinent data. It is due to regional effect. This study is mainly based on HKSAR regional data. Ecovinent data is for counter-check purpose.

Table 5.10 Mass weight of each component inside Air Handling Unit

Section	Component	Material	Size	Mass (kg)
Frame and Panel Section	Corner	Rubber	40mm (W) x 5 mm (Thickness) x 50 mm (L)	0.09
	Double skin type Panel	Polyurethane layer (density>45)		
			Top: 830mm x 1110 mm x 8.8 mm	0.24
			Bottom: 830mm x 1110 mm x 8.8 mm	0.24
			Left: 1110 mm x 560 mm x 8 mm (need to exclude pipe area)	0.16
			Right: 1110 mm x 560 mm x 8 mm	0.16
			Front: 830 mm x 560 mm x 8 mm (need to exclude duct area)	0.11

			No panel at the back	0.00
		Blue Color Painted Steel		
			Top: 830mm x 1110 mm x 8.8 mm	8.62
			Bottom: 830mm x 1110 mm x 8.8 mm	8.62
			Left: 1110 mm x 560 mm x 8 mm (need to exclude pipe area)	5.80
			Right: 1110 mm x 560 mm x 8 mm	5.82
			Front: 830 mm x 560 mm x 8 mm (need to exclude duct area)	3.90
			No panel at the back	0.00
		Aluminum Frame		
			Top: 830mm x 1110 mm x 8.8 mm	1.48
			Bottom: 830mm x 1110 mm x 8.8 mm	1.48
			Left: 1110 mm x 560 mm x 8 mm (need to exclude pipe area)	1.56
			Right: 1110 mm x 560 mm x 8 mm	1.56
			Front: 830 mm x 560 mm x 8 mm (need to exclude duct area)	3.74
			No panel at the back	0.00
	Panel Gasket	Rubber (40mm thickness)		3.68
	10 nos. M8 Screw for motor, fan and panel handles installation	Steel		
				0.10
				0.15
	10 nos. Washers for motor, fan and panel handles installation			10.71
				2.69
	Electrical Wire	Copper	Core Area $1.5mm^2$, length:6.65m	0.09
		PVC	length:6.65m insulation thickness = 1mm	0.27
	Insulation	fiber glass (chiller water pipe)	Pipe dia = 25.4 mm, thickness 13mm, total length = 13.38m	0.42
		Aluminum Foil	Pipe dia = 25.4 mm, fiber glass thickness = 13 mm, thickness = 0.1mm, length = 8.58m	0.10
	6 handles	rubber	160mm (L) x 25mm (W) x 10mm (D)	0.36
	16 locks		10mm (L) x 5mm (W) x 5mm (D)	0.01
	7nos. M8 Screw for Framework for mounting AHU	Galvanized Iron	thickness=3mm width=40mm extra length for screw connection =8cm	16.42
			4	0.09
Motor	(7-7 x 1 No.) Fan	Hot-dip	1.1 kW	76.89

and Fan Section		galvanized steel sheets		
	Fan wheel (Bell Wheel SPA80-2-1210 (20,6,22.5))	Cast Iron	---	1.29
	Motor (TEFC) (1.1KW x1No.)	stainless steel		0.08
		stainless steel		9.82
		Copper		6.74
		cast iron		4.51
		stainless steel		0.21
	Motor wheel (Bell Wheel SPA112-2-1610(24,8,27))	Cast Iron		8.70
	ST-40 Spring x 4	Galvanized Steel	Diameter 30mm thickness 2mm number of rounds 4	0.04
	bell	rubber	1000 mm (L) x 20 mm (W) x 2 mm (D)	0.06
Coil Section	Coil (3CW4/12H-400x610A x 1No) with connection points	black steel	25.4 mm (Dia) x 3.14 x 4290 mm (L) x 0.2 mm (Thickness)	0.54
		black steel	25.4 mm (Dia) x 3.14 x 4460 mm (L) x 0.2 mm (Thickness)	0.56
		cast iron	25.4 mm (Dia) x 3.14 x 4630 mm (L) x 0.2 mm (Thickness)	0.59
		Copper	12.7 mm x 3.14 x 400 mm (L) x 0.15 mm (Thickness), Number of tube rows : 4, 8 parallel tubes in each row	1.23
	Exchange Tubes	Aluminum	0.1524 mm (W) x 140 (L) x 400 (H) x 284 (No. of fins = 610/(0.1524 +2)	6.54
	Fins	Copper	(((22+1.5+1.5)/2)^2-(22/2)^2)*3.14*400*4/(1000^3), thickness = 1.5 mm, Diameter 22mm	3.81
	Header (header in)	copper	(((22+1.5+1.5)/2)^2-(22/2)^2)*3.14*400*4/(1000^3)	3.81
	suction (header out)	Galvanized Iron	980mm (L) x 860mm (W) x 1mm (thickness)	6.63
	condensing panel	Aluminum	670(L) x 460 (H) x 140 (D)	21.64
	Coil Frame	Aluminum	46 x 455 x 400	8.91
Filter Section	Filter Section (46mm D455H x 400W x2Nos.)	Galvanized Iron	80 mm (H) x 900 mm (D) x 25 mm (Thickness)	28.33
	Slide in guides for filter		Total Mass	286.42

Table 5.11 are listed out the energy Intensity data from LCI & Ecovinent for the related material consisted inside an air handling unit, as well as each component of various ventilation systems. The embodied energy of the Polyurethane is unavailable in the LCI database. It is assumed that the value is the same as the embodied energy in the Econivent database. Using rubber is an example to support this assumption in Table 5.11, it is found that the embodied energy of 101.7 MJ/Kg and embodied carbon of 3.18 CO_2/kg in rubber obtained from Ecoivent data base are similar to HKSAR data, which are 100.96 MJ/kg and embodied carbon of 2.66 CO_2/kg. These are involving all process from cradle to gate. Details breakdown of embodied energy and carbon for rubber, retrieved from Ecoinvent's life cycle inventory are tabulated in Table 5.12 and 5.13 to express all concern items.

Table 5.11 Energy Intensity data (LCI & Ecovinent)

	LCI		Ecoinvent	
Material	Embodied Energy	Embodied Carbon	Embodied Energy	Embodied Carbon
	MJ/kg	CO_2/kg	MJ/kg	CO_2/kg
Aluminum	211.00	18.10	205.26	7.07
Copper	22.60	1.92	47.58	1.91
Fiber Glass	31.70	2.47	27.50	1.99
Iron	14.30	0.83	27.50	1.99
Polyurethane	72.10	3.76	72.10	3.76
PVC	87.90	2.17	80.04	2.01
Rubber	*101.70*	*3.18*	*100.96*	*2.66*
Steel	52.70	3.43	38.00	1.72

Table 5.12 Embodied energy of Rubber

Resources of energy input	By-production energy input	Resource input per unit of rubber (MJ/unit)	Energy per unit of resource input (MJ/unit)	Quantity consumed (1kg)	Cumulative energy (MJ)/(KG)
[Resources from biosphere]	Energy, gross calorific value, in biomass	0.9624	1	1	0.9624
[Resources from biosphere]	Energy, gross calorific value, in biomass, primary forest	0.00004038	1	1	0.00004038
[Renewable energy resources from air]	primary energy from wind power	0.115	1	1	0.115
[Renewable energy resources from water]	primary energy from hydro power	0.9592	1	1	0.9592
[Renewable energy resources from air]	primary energy from solar energy	0.001657	1	1	0.001657
[Non-renewable energy resources from ground]	brown coal; 11.9 MJ/kg	11.9	0.2811	1	3.34509
[Non-renewable energy resources from ground]	hard coal; 26.3 MJ/kg	26.3	0.2042	1	5.37046
[Non-renewable energy resources from ground]	crude oil; 42.3 MJ/kg	42.3	1.311	1	55.4553
[Non-renewable energy resources from ground]	uranium (0.03*)	0.00001411	82100000	1	34.75293
		Sum of Cumulative energy, MJ/Kg			**100.9620774**

Table 5.13 Embodied carbon of Rubber

Area of Emission	Emission product	Emission/unit of product output (kg)	CO_2 equivalent factor	Total mass of CO_2/unit of output
[Emissions to urban air close to ground]	carbon dioxide (biogenic)	0.03166	1	0.03166
[Emissions to non-urban air or from high stacks]	carbon dioxide (biogenic)	0.00343	1	0.00343
[Emissions to air, unspecified]	carbon dioxide (biogenic)	0.001409	1	0.001409
[Emissions to urban air close to ground]	carbon dioxide (fossil)	1.726	1	1.726
[Emissions to non-urban air or from high stacks]	carbon dioxide (fossil)	0.596	1	0.596
[Emissions to lower stratosphere and upper troposphere]	carbon dioxide (fossil)	5.478E-07	1	5.478E-07
[Emissions to air, unspecified]	carbon dioxide (fossil)	0.09779	1	0.09779
[Emissions to non-urban air or from high stacks]	carbon dioxide	0.00006249	1	0.00006249
[Emissions to urban air close to ground]	methane (biogenic)	0.00002909	23	0.00066907
[Emissions to non-urban air or from high stacks]	methane (biogenic)	0.00001054	23	0.00024242
[Emissions to air, unspecified]	methane (biogenic)	0.00002565	23	0.00058995
[Emissions to urban air close to ground]	methane (fossil)	0.004873	23	0.112079
[Emissions to non-urban air or from high stacks]	methane (fossil)	0.002525	23	0.058075
[Emissions to lower stratosphere and upper troposphere]	methane (fossil)	8.696E-12	23	2.00008E-10
[Emissions to air, unspecified]	methane (fossil)	0.00003989	23	0.00091747
[Emissions to urban air close to ground]	nitrous oxide	0.00009667	296	0.02861432
[Emissions to non-urban air or from high stacks]	nitrous oxide	0.00001136	296	0.00336256
[Emissions to lower stratosphere and upper troposphere]	nitrous oxide	5.217E-12	296	1.54423E-09
[Emissions to air, unspecified]	nitrous oxide	0.00001042	296	0.00308432
			Total CO_2 (eqv) (Kg)	**2.66398615**

The database developed by Electrical and Mechanical Services Department of the HKSAR government with LCI data is used for estimating the cumulative energy of various ventilation systems. Table 5.14 is presented the breakdown of Cumulative energy (LCI data) of the air handling unit (AHU) installing inside the environmental chamber. Each assemble component are based on overview of ductwork and diffuser use for mixing, displacement, stratum ventilation as illustrated in Figure 5.4 to 5.6.

The calculation has considered the amount of material in different sizes in each section of AHU. The numbers of each assemble component are determined in accordance with the manufacturer catalogue and installation guideline. The maximum spacing of each supporting fixing to the ceiling or wall or on the floor is highlighted in the specification document. The embodied energy of the each connection accessories and the valves are estimated by their mass content times the embodied energy value per unit mass of copper in the EMSD. The embodied energy inventories in this database are in the "Cradle to-as-built" stage. All the values have included all the processes across the entire (LCA) boundary. The total cumulative energy for the air handling unit is 18,823 MJ/Kg and 1,407 CO_2/kg. Other components, such as ductwork, dampers, and terminals required for mixing, displacement and stratum ventilation system are indicated on Table 5.14.

Table 5.14 Cumulative energy (LCI data) of the air handling unit (AHU)

Chamber	Description of each assemble component	Material	Embodied energy (MJ) [Mass, Kg x Energy intensity, MJ/Kg]	Embodied Carbon (CO_2/kg)
Enclosure	Corner	Rubber	9.15	0.29
	Double skin type Panel	Polyurethane layer (density>45)		
		Top	17.54	0.91
		Bottom	17.54	0.91
		Left (connect to water pipes)	11.83	0.62
		right	11.83	0.62
		front (supply)	7.93	0.41
		Blue Color Painted Steel		
		Top	454.45	29.58
		Bottom	454.45	29.58
		Left (connect to water pipes)	305.87	19.91
		right	306.62	19.96
		front (supply)	205.47	13.37
		Aluminum Frame		
		Top	324.03	26.78
		Bottom	324.03	26.78
		Left (connect to water pipes)	341.18	28.20
		right	341.18	28.20
		front (supply)	820.13	67.78
	Panel Gasket	Rubber	374.21	11.70
	M8 Screw	Steel		
		Motor: 4	5.27	0.34
		Fan : 6	7.91	0.51
	Washer	Steel		
		Motor: 4	564.18	36.72
		Fan : 6	141.95	9.24
	Electrical Wire	Copper	2.02	0.17
		PVC	24.15	0.60
	Insulation	fiber glass (chiller water pipe)	13.34	1.04
		Aluminum Foil	20.93	1.73
	6 handles	rubber	36.61	1.14
	16 locks	rubber	0.61	0.02
	Framework for mounting AHU			
	Framework	Galvanized Iron	234.80	13.56
	M8 Screw x 7		1.23	0.07

Supply Coil	(7-7 x 1 No.) Fan	Hot-dip galvanized steel sheets	4052.30	263.75
	Fan wheel (Bell Wheel SPA80-2-1210 (20,6,22.5))	Cast Iron	18.41	1.06
	Motor (TEFC) (1.1KW x1No.)			
	Motor wheel (Bell Wheel SPA112-2-1610(24,8,27))	Cast Iron	124.47	7.19
	Bearing	stainless steel	4.37	0.28
	Rotor	stainless steel	517.38	33.67
	Stator	Copper	152.27	12.94
	Controller	cast iron	64.52	3.73
	shaft	stainless steel	11.33	0.74
	ST-40 Spring x 4	Galvanized Steel	2.08	0.14
	bell	rubber	6.10	0.19
	chilled water connection pipe in	black steel	28.36	1.85
	chilled water connection pipe out	black steel	29.49	1.92
	condensing water drain pipe	cast iron	8.38	0.48
	Exchange Tubes	Copper	27.76	2.36
	Fins	Aluminum	1380.82	118.45
	Header (header in)	Copper	34.44	2.93
	suction (header out)	copper	34.44	2.93
	condensing panel	Galvanized Iron	94.85	5.48
	Coil Frame	Aluminum	4566.21	391.70
Filter	Filter Section (46mm D455H x 400W x2Nos.)	Aluminum	1879.51	161.23
	Slide in guides for filter	Galvanized Iron	405.15	23.40
		Total Cumulative value:	**18823.07**	**1407.15**

Table 5.14 is presented the embodied energy of each equipment/material/parts from factory to site installation of mixing, displacement and stratum ventilation methods. The components of various systems listed in the above tables are manufactured from industrial region in Guangdong province of China. The vehicle travel distance between Dongguang to HKSAR is about 150 km as shown in Table 5.15. From the Chinese Life Cycle Database

stored inside “eBalance”, the embodied energy for an 8 tons lorry is 2.474 MJ per ton-kilometer.

Table 5.15 Transportation distance to HKSAR from different countries

Country	Sea distance to Hong Kong (km)	Inland Distance (km)
Australia	7152.0	4946.3
Belgium	16343.3	311.8
Brazil	20840.0	5206.6
Canada	15839.0	5636.6
China (Guangdong)	150.0	250.0
France	16115.0	1325.4
Germany	18472.0	1066.3
India	6797.0	3077.9
Indonesia	3311.0	2462.8
Italy	13400.0	979.5
Japan	3155.0	1097.2
Korea	2246.7	561.5
Malaysia	2122.0	104.5
Singapore	2496.7	4511.1
Spain	13571.7	1267.9
Taiwan	632.0	107.0
Thailand	2576.7	1279.4
UK	18240.0	877.9
USA	18753.3	5463.4
Vietnam	1178.0	1024.5

Table 5.16 Cumulative energy (LCI Data) of installation for Mixing, Displacement and Stratum methods

Ventilation Methods	Description of Major Components	Mass (kg)	Embodied energy (MJ) [Mass, Kg x Energy intensity, MJ/Kg]	Embodied Carbon (CO_2/kg)
Air Handling Unit for various ventilation		286.42 *(from Table 5.10)*	18,823.07 *(from Table 5.14)*	1,407.15 *(from Table 5.14)*
Mixing (Base)	Ductwork , 14.93 m² x 2mm thickness of Galvanized iron	234.998	3,360.47	194.11
	Insulation, 14.93 m² x 13mm thickness of Fibre Glass	93.1632	2,953.27	230.11
	Insulation for Ductwork, 14.93 m² x 0.1mm thickness of Aluminum Foil	4.0311	850.56	72.96

	Supply & Return Diffuser, 3.24 m² x 3 mm thickness of Aluminum	26.244	5,537.48	475.02
Cumulative Energy of Mixing Ventilation:		623	**31,524.86 (Base)**	**2,379.35 (Base)**
Displacement	Ductwork , 41.85 m² x 2mm thickness of Galvanized iron	658.719	9,419.68	544.10
	Insulation, 41.85 m² x 13mm thickness of Fibre Glass	261.144	8,278.26	645.03
	Insulation for Ductwork, 41.85 m² x 0.1mm thickness of Aluminum Foil	11.2995	2,384.19	204.52
	Supply & Return Diffuser, 1.8 m² x 3 mm thickness of Aluminum	14.58	3,076.38	263.90
Cumulative Energy of Displacement Ventilation:		1210	**41,981.59 (-33.17%)**	**3,064.69 (-28.8%)**
Stratum Ventilation	Ductwork , 43.12 m² x 2mm thickness of Galvanized iron	678.709	9,705.54	560.61
	Insulation, 43.12 m² x 13mm thickness of Fibre Glass	269.069	8,529.48	664.60
	Insulation for Ductwork, 43.12 m² x 0.1mm thickness of Aluminum Foil	11.6424	2,456.55	210.73
	Supply & Return Diffuser, 0.72 m² x 3 mm thickness of Aluminum	5.832	1,230.55	105.56
Cumulative Energy of Stratum Ventilation:		1231	**40,745.19 (-29.25%)**	**2,948.65 (-23.93)**

As presented in Table 5.15, an actual Figure of embodied energy may be varied due to regional differences. Since the materials of each component in various ventilation methods are made in same country of China and then transport into the site for construction. Thus, this factor is ignored.

Table 5.17 LCA cumulative energy of the equipment/material/parts transportation

Transportation Method by 8 Tons Lorry.	Total mass of ventilation system (kg or Ton)	Distance (km)	Embodied Energy factor (MJ/Ton.km)	Embodied Energy (MJ)
Mixing	623 or 0.623	150	2.474	231
Displacement	1210 or 1.21	150	2.474	449
Stratum	1231 or 1.231	150	2.474	457

By summing up the embodied energy of all components from Table 5.15, 5.16 & 5.17, the total embodied energy used for equipment/material/parts including transportation from factory gate to site installation are summarized in Table 5.18.

Table 5.18 LCA cumulative total energy in supply and installation phase for three ventilation systems

Ventilation method	Cumulative energy in AHU (Table 5.15) + Transportation (Table5.16) + Accessories (Table5.17), MJ	Total Cumulative embodied energy, MJ	Total Cumulative embodied energy, kWh (=3.6 x MJ)
Mixing	18,823+31,525+231	50,579	182,084
Displacement	18,823+41,982+449	61,254	220,514
Stratum	18,823+40,745+457	60,025	216,090

5.4.2.2 Use-phase assessment.

The use-phase assessment of the study consists of GHGs caused by energy consumption and maintenance activities of the system. The primary data used in estimating the energy use related GHG emissions are the estimated energy consumption figures for each ventilation system based on territory-wide emission factor of GHG coming from utility power generation is 0.7 kg CO_2-eq/kWh. From Table 3.10b: Cooling sensible energy for a space conditioned by various ventilation methods, the year-round total primary energy consumption of 64,737kWh; 62,198 kWh; 54,475 kWh are used in use-phase assessment and are tabulated in Table 5.19. As result, the operating energy consumption for mixing, displacement and stratum ventilation systems can be estimated by the year-round total energy consumption times the concerned service years and emission factor of GHG. The estimated annual GHG emission of three ventilation system analysis based on same operating basis are tabulated in Table 5.20. The GHG emission rate combined with (1) supply and installation phase and, (2) user phase of displacement and stratum ventilation during LCA for 20 services year are less than mixing ventilation around 3.67% and 15.51% respectively. At the end of 20 services year, life cycle resulted in GHG emissions of 12,815,911 kg-CO_2 (base) for the mixing ventilation option, 12,345,168 kg-CO_2 (3.67% less) for the displacement ventilation option, 10,828,363 kg-CO_2 (15.51% less) for stratum ventilation.

Table 5.19 GHG Estimation in user assessment phase for three ventilation systems in 5,10,15,20 service year

ventilation systems	Year-round total primary energy consumption (kWh)	User assessment phase - Operation Energy Consumption for different service years, Kg-CO_2 (KWh x 0.7 kg CO_2-eq/kWh x service years)			
		5	10	15	20
Mixing Ventilation:	64,737.00	226,580	453,159	679,739	906,318
	Base				
Displacement Ventilation:	62,198.00	217,693	435,386	653,079	870,772
	-4%	-3.92%	-3.92%	-3.92%	-3.92%
Stratum Ventilation:	54,475.00	190,663	381,325	571,988	762,650
	-16%	-15.85%	-15.85%	-15.85%	-15.85%

Table 5.20 GHG Estimation combined with two phases for three ventilation systems in 5,10,15,20 service year (% difference is based on mixing ventilation)

	GHG Estimation (Kg-CO_2) of LCA for Yr5, Yr10, Yr15 & Yr20 Combined with [1] Supply and Installation Phase, and [2] User assessment phase			
System Life Time (Year)	5	10	15	20
Mixing Ventilation (Reference Case)	354,039	3,299,572	7,264,713	12,815,911
Displacement Ventilation	372,053 (5.09%)	3,202,062 (-2.96%)	7,011,690 (-3.48%)	12,345,168 (-3.67%)
Stratum Ventilation	341,926 (-3.42%)	2,820,538 (-14.52%)	6,157,132 (-15.25%)	10,828,363 (-15.51%)

Since the reduction of GHG and embodied energy are found in both of displacement & stratum ventilation after 10 service year, in comparing with mixing ventilation. No need to further estimate the energy and greenhouse gas payback period of the displacement and stratum ventilation in this case.

5.4.2.3 Further study in HKSAR

The carbon intensity of the energy building system is likely to decrease in the future along with technical advancements in energy building system processes. This would significantly decrease the importance of GHG emissions occurring in the distant future. However, the carbon payback times of the different energy options of the study are only a few years, so future decreases in energy building system intensities will not change the mutual order of the options, although the GHG emissions of the building system solutions decrease most radically in the future due to a combination of high energy consumption and lower energy generation intensity in the future.

EMSD’s Technical Taskforce has focus on developing the building energy efficiency ordinance with respect to the latest development of the relevant technology and practices in other regions. Two mandatory requirements for HKSAR commercial buildings has been formulated and stated in the new Building Energy Code (BEC) and Energy Audit Code (EAC) under current Buildings Energy Efficiency Ordinance. The Code provides useful information for the building owner to decide on and implement the energy saving measures for environmental consideration and economic benefits. Energy audit can achieve energy efficiency and conservation through the implementation of energy management opportunity (EMO) identified in the audit. Besides, government has established guidelines for a comparative methodology framework for calculating cost-optimal designs. In this framework, there is a cost group labelled cost of greenhouse gas emissions. This cost group implies that in the future, there may be costs for generating greenhouse gases. If costs for producing greenhouse gas emissions would be applied, this would further enhance the connection between Life Cycle Cost (LCC) and Life Cycle Assessment (LCA). In order to execute an LCC model, an LCA would have to be further executed to obtain the data for the costs of greenhouse gas emissions.

5.5 Concluding remarks

This cost effectiveness study analyses LCC (life cycle costs) and carbon emissions of displacement and stratum ventilation systems, in comparison with conventional mixing ventilation system as reference case, through a methodological life cycle framework in term of 5, 10, 15 and 20 service years, which are summarized in Table 5.21 and Table 5.22 for easy reference. Results of these tables are extracted from data in Table 5.9, 5.18, 5.19 & 5.20.

Table 5.21 Cumulative energy in all phases for LCC analysis

Ventilation mode	Supply and installation phase (from Table 5.18)		User assessment phase (from Table 5.19)	[4] = [2] +[3] Total cumulative energy in all phases during first year operation, kWh
	[1] Total Cumulative embodied energy, MJ	[2] Total Cumulative embodied energy, kWh =3.6 x MJ	[3] Year-round total primary energy consumption ,kWh	
Mixing Ventilation (Reference Case)	50,579	182,084	64,737	246,821
Displacement Ventilation	61,254	220,514	62,198	282,712 (14.54%)
Stratum Ventilation	60,025	216,090	54,575	270,665 (9.66%)

The total cumulative energy of all phases in three ventilation methods is summarized in the above Table 5.21. The related GHG emissions are the estimated energy consumption figures for each ventilation system based on territory-wide emission factor of GHG coming from utility power generation is 0.7 kg CO_2-eq/kWh. As a result, the GHG emission rate combined with (1) supply and installation phase and, (2) user phase of displacement and stratum ventilation during LCA for 20 services year are summarized in Table 5.22. It was found that although the total accumulated energy during the first year operation of mixing is lower than displacement and stratum ventilation, as shown in Table 5.21, the significant GHG emission can be reduced due to less annual energy consumption at the end of 20 services year.

In Table 5.22, life cycle resulted in GHG emissions of 12,815,911 kg-CO_2 (base) for the mixing ventilation option, 12,345,168 kg-CO_2 (3.67% less) for the displacement ventilation option, 10,828,363 kg-CO_2 (15.51% less) for stratum ventilation. Cost reduction of stratum ventilation in comparing with mixing ventilation are found as HK$477,371 (6.58% less), HK$685,843 (7.37% less), HK$779,630 (7.75% less), and HK$823,456 (7.94% less) in 5, 10, 15 and 20 service years. This implies that a payback period of stratum ventilation have been feasibly achieved by extending its life span through better maintenance work.

Table 5.22 Summary of LCC and GHG analysis in service year at 5, 10, 15 & 20

Ventilation mode	System Life Time (Year)			
	5	10	15	20
[A] Net present value, HK$ (from Table 5.9)				
Mixing Ventilation (Reference Case)	510,999	740,386	845,105	894,517
Displacement Ventilation	512,568 0.31%	740,006 -0.05%	843,203 -0.23%	891,702 -0.31%
Stratum Ventilation	**477,371 -6.58%**	**685,843 -7.37%**	**779,630 -7.75%**	**823,456 -7.94%**
[B] User assessment phase - Operation Energy Consumption, Kg-CO_2 (from Table 5.19)				
Mixing Ventilation (Reference Case)	226,580	453,159	679,739	**906,318**
Displacement Ventilation	217,693 -3.92%	435,386 -3.92%	653,079 -3.92%	**870,772 -3.92%**
Stratum Ventilation	190,663 -15.85%	381,325 -15.85%	571,988 -15.85%	**762,650 -15.85%**
[C] Greenhouse gas, GHG Estimation, combined with all phases including supply, installation and user assessment phase, Kg-CO_2 (from Table 5.20)				
Mixing Ventilation (Reference Case)	354,039	3,299,572	7,264,713	**12,815,911**
Displacement Ventilation	372,053 5.09%	3,202,062 -2.96%	7,011,690 -3.48%	**12,345,168 -3.67**
Stratum Ventilation	341,926 -3.42%	2,820,538 -14.52%	6,157,132 -15.25%	**10,828,363 -15.51%**

CHAPTER 6: CONCLUSION AND FURTHER WORK

The entire research works of "A Study of Thermal Comfort and Cost Effectiveness of Stratum Ventilation" and major contributions of this research are highlighted as below. The possible further investigations are also explored.

6.1 Summary of major contributions and findings

Neutral temperatures have been estimated for mixing, displacement and stratum air distribution strategies. The actual cost effectiveness of the installed ventilation systems has also been investigated by life cycle assessment. These results are quantified together with mixing and displacement ventilation as reference cases inside a full-scale newly developed environmental chamber. The result provides a scientific basis for adopting stratum-ventilated medium-size air conditioned space in subtropical climates. Both of new-build and retrofit space in small-medium size are applicable to be used by practitioners.

6.1.1 Environmental chamber development

Full-scale environmental chamber of 8.8 m (length) × 6.1 m (width) × 2.4 m (height) with data collection equipment/device has been developed and set up to conduct all human comfort test and to collect experimental data for analysing thermal comfort and cost effectiveness of the mixing, displacement and stratum ventilation systems. The installation detail of the chamber has been illustrated in Figure 2.3 in Chapter 2. Computational fluid dynamic simulation has been used to assist and finalise the necessary information for establishing this setup, such as the supply air parameters, heat sources identification and furniture arrangement. The model is illustrated in Figure 2.4. Through building management system (BMS), all operating equipment/devices, such as air handling unit with variable speed of supply fan, CO_2 sensor for fresh air control, air distribution device and water distribution device have been monitored and controlled during all human comfort tests and energy consumption study. Apart from the BMS system, five measurement points at 1.1 m above finishing floor level (AFFL) have also been set up within breathing zone during human comfort test as shown in Figure 2.12.

6.1.2 Neutral temperature estimation

Neutral temperature of mixing, displacement, stratum air distribution methods supported by human comfort test and experimental data have been found. The neutral temperatures of the younger under the mode of mixing, displacement, stratum, modified-stratum-1, modified-stratum-2, and modified-stratum-3 have been identified as 24.6℃, 25.1℃, 25.6℃, 26.0℃, 27.1℃ and 27.3℃ at 10 air change per hour (ACH) respectively and become 24.8℃, 25.3℃, 26.6℃, 27.4℃, and 27.9℃ at 15 ACH respectively. The younger is between 20 and 23 years old, their clothing insulation and actively level are 0.57 clo and 1.0 met respectively. More energy can be conserved in the modified-stratum-3 mode with satisfactory thermal comfort level at elevated air conditions. The distribution of thermal sensation vote has been illustrated in previous Table 3.4.

These findings for the conventional mixing ventilation modes are similar to the finding by Mui and Wong (2007) with the neutral temperatures being 24.9℃ as evaluated from 422 responded occupants based on the thermal environment with the mean air speed under 0.1 m/s, a clothing level of 0.57 clo and a metabolic rate of 1 met for a sedentary working environment.

The highest acceptability for thermal comfort at 95.8% with 10 ACH using the human subject tests have been found at the following six conditions: mixing ventilation at 24℃, displacement ventilation at 24℃, stratum ventilation at 24℃, modified-stratum-1 ventilation at 24℃ & 26℃, modified-stratum-2 ventilation at 26℃, and modified-stratum-3 ventilation at 28℃. The highest acceptance percentage can be elevated to 28℃ with air flow supply increased to 15 ACH under modified-stratum-2 and modified-stratum-3 only. In term of increasing the air flow supply rate from 10 to 15 ACH, the neutral temperatures for both the mixing ventilation and displacement ventilation modes have been raised by 0.2℃. By supplying 50% more air from 10 ACH to 15 ACH, stratum ventilation probably allows subjects to accept an additional 0.2℃ to 0.6℃ as compared to an additional 0.2℃ under mixing ventilation and displacement ventilation modes. Therefore, air speed is a control variable that can enhance human thermal comfort.

6.1.3 Uniformity of stratum ventilation study

Three sets of collected data from three individual investigators in three consequent summer times have been used to support the significant level of the uniformity in thermal environment of stratum ventilated room. In most of the cases studied, the values of air distribution performance index (ADPI) derived from measured data are not less than 80%. This result shows that thermal comfort condition in the occupied zone is uniform. The results also indicate that the standard deviations of thermal sensations of subjects in thermal environments with stratum ventilation are comparable to those with mixing ventilation. This demonstrates that the thermal conditions in stratum ventilation are uniformly distributed.

This study also experimentally confirmed that stratum ventilation could realise thermal comfort under higher nominal room temperatures. Therefore, it is easier to meet the requirements of elevated room temperatures recently imposed by East Asia countries. By comparing actual mean thermal sensations (ATS) with the value of predicted mean vote (*PMV)* small deviations have been found. *PMV* model could therefore be used to predict thermal sensation in stratum ventilated rooms. Both objective experimental measurements and subjective human comfort tests confirm that the thermal environment in the occupied zone of a stratum ventilated room is uniform.

6.1.4 Cost effectiveness analysis

This analysis covers the stage of "cradle to as-built" and annual sensible energy conservation of air-conditioning equipment/device. This cost effectiveness study involved analysis of life cycle cost and carbon emissions of mixing, displacement and stratum ventilation systems in terms of 5, 10, 15 and 20 service years.

The trend of descending life cycle cost for the three ventilation systems is displacement, stratum and mixing. Displacement ventilation is the most expensive one in comparison with mixing and stratum ventilation as shown in Figure 5.8. Table 5.9 presented the percentages of cost reduction of stratum ventilation in comparing with mixing ventilation which measure 6.58%, 7.37%, 7.75%, and 7.94% in 5, 10, 15 and 20 service years respectively.

GHG emission rate combined with supply & installation phase and user phase during LCA for 20 services year are 12,815,911 kg-CO_2 (base) for the mixing ventilation, 12,345,168 kg-CO_2 (3.67% less) for the displacement ventilation and 10,828,363 kg-CO_2 (15.51% less) for stratum ventilation as shown in Table 5.20. The stratum ventilation is the best option for both of cost reduction and less carbon emission based on 5, 10, 15 & 20-year time horizon.

6.2 Contributions of the research

The estimated neutral temperature of stratum ventilation induced a number of implications for thermal comfort in air-conditioned space and energy consumption of the associated air-conditioning system. With respect to the thermal comfort aspect, it provides scientific basis for the feasibility of elevated room temperature not only for stratum, but also for mixing and displacement ventilation systems. For the energy consumption aspect, it shows considerable potential for energy saving while adopting this set-point during actual operation.

The confirmation of the uniform thermal environment served by stratum ventilation allows the design of thermally-comfort environment by using stratum air distribution strategy. It enables large extension on adopting conventional methodology for air distribution design.

Through life cycle assessment of a full-scale of mixing, displacement, stratum ventilation systems, their corresponding sensible energy consumption and greenhouse gas emission have been found. These favourable results from energy saving and carbon emission reduction can re-confirm the advantages of stratum ventilation in air conditional medium-size area.

6.3 Recommendations for future work

Further studies are required to investigate the effects of the thermal sensation perceived by human subject with different conditions, such as body mass index, clothing value, age, and gender. Correlation between the air velocity, air temperature, acceptability, and achieved thermal comfort level can be explored. Moreover, the comparison between any new criteria as stated in the latest version of ASHARE standard 55-2013 with my estimated acceptable percentage are collected from all human comfort tests in six ventilation modes.

Due to the limitation of steady states and simply comparison with sensible space load used in the study, more comprehensive energy saving potential can be estimated. Higher ventilation efficiency of stratum ventilation, reduced ventilation rates, and latent load estimation during part-load condition should be considered. Besides, fresh air may be reduced due to higher ventilation efficiency in comparison with mixing ventilation and less fan power due to shorter ductwork and less requirement of external static pressure in comparison with displacement ventilation. More comprehensive study of all year-round energy profile should be conducted to improve the accuracy of cost effectiveness analysis in overall energy cost saving by stratum ventilation in comparison with mixing and displacement ventilation systems by simulation model, such as TRNSYS. Moreover, more energy can be saved if further investigation for lengths of the pull down periods can be performed. The pull down period is the time used to achieve a comfortable thermal environment. For example, to estimate how much energy use in early morning while an air conditioning system is started to cool down a non-conditioned area in order to achieve an interior thermal comfort environment.

During the life cycle assessment, it is difficult to estimate the embodied carbon accurately in the absence of an agreed framework to assess carbon emissions of construction materials from different supply regions at product level of “cradle to as-built”, which include cradle to gate, gate to gate and gate to site. In pursuit of low carbon construction of the air distribution system, the fundamental issues on how to assess the carbon emission of air distribution systems need to be further investigated. Carbon embodied in a construction facility at supply phase, “cradle-to-gate” is an important but always neglected topic. The assessment framework serves to provide specific requirements according to the production process of various imported air distribution systems as well as that stipulated in the existing local and international best practices.

REFERENCES

Abanto, J., Barrero, D., Rggio, M. and Ozell., B. (2004). Airflow modeling in a computer room. Building and Environment, 39 (12), 1393-1402.

ANSI/ASHRAE Standard 55 (2010). Thermal environmental conditions for human occupancy.

Ardente, F., Beccali, G., Cellura, M., Brano, V.L. (2005). Life cycle assessment of a solar thermal collector. Renewable Energy,30,1031-1054.

Arens, E., Xu, T., Miura, K., Zhang, H., Fountain, M., Bauman, F. (1998). A study of occupant cooling by personally controlled air movement. Energy and Buildings, 27(1), 45–59.

Ariff, M.,Salim, S.M., Cheah,S.C.(2009). Wall y+ approach for dealing with turbulent flow over a surface mounted cube: Part 2- high Reynolds number. Seventh International Conference on CFD in the Minerals and process Industries, Melbourne, Australia, December 2009.

Arens, E., Turner, S., H. Zhang,H., Paliaga G. (2009). Moving air for comfort. ASHRAE Journal, 51 (5): 18-28.

ASHRAE: American Society of Heating, Ventilation, and Refrigerating Engineers (2013). ASHRAE Handbook: Fundamentals: SI Edition. Atlanta, Georgia, USA

ASHRAE 62-2013. ASHRAE Standard 62.1 (2013): Ventilation for acceptable indoor air quality, Atlanta, Georgia, USA, American Society of Heating, Refrigerating, and Air-conditioning Engineers, Inc.

.

ASHRAE 55-2010. ASHRAE Standard 55 (2010): Thermal environmental conditions for human occupancy, Atlanta, American Society of Heating, Refrigerating, and Air-conditioning Engineers, Inc.

Athena Sustainable Materials Institute – LCI databases. Retrieved August 2014 from http://www.athenasmi.org/our-software-data/lca-databases/

Awbi H.B. (2011). Energy efficient ventilation for retrofit buildings. Proceedings of 48th AICARR international conference on energy performance of existing buildings, Baveno, Italy. September 2011, 23-46.

Awbi H.B. (2000). Air distribution in rooms: ventilation for health and sustainable environment. Elsevier.

Awbi, H.B. (1996).Thermal Boundary Conditions for internal room Surfaces. ROOMVENT'96, 1, 243-250.

Balazova, I., Clausen, G., Rindel, J.H., Poulsen, T., Wyonm D.P. (2008). Open-plan office environments: a laboratory experiment to examine the effect of office noise and temperature on human perception, comfort and office work performance. 11th International Conference on Indoor Air Quality and Climate, Copenhagen, Denmark August 2008.

Baturin, V.V. (1972). Fundamental of industrial ventilation, 3rd England ed. Translated by O.M. Blunn. Pergamon Press, New York.

BEES, **B**uilding for **E**nvironmental and **E**conomic **S**ustainability. Retrieved from June 2014 from http://www.nist.gov/el/economics/BEESSoftware.cfm

Bolashikov, Z., Nikolaev, L., Melikov, A., Kaczmarczyk, J. and Fanger, P.O. (2003). New for terminal devices with high efficiency for personalized ventilation application. Proceedings of Healthy Buildings, 2, 580-855.

Brown, G.A. and Williams, G.M. (1982). The effect of head cooling on deep body temperature and thermal comfort in man, Aviat. Space Environ. Med., 53, 583–586.

Brohus, H. and Nielsen, PV. (1996). Personal exposure in displacement ventilated rooms. Indoor Air, 6,157-167.

BS European Standard EN ISO 7730-2005. Ergonomics of the thermal environment – Analytical determination and interpretation of thermal comfort using calculation of the *PMV* and *PPD* indicates and local thermal comfort criteria. Geneva: International Organization for Standardization; 2005.

Candido, C., de Dear, R.J., Lamberts, R., Bittencourt, L. (2009). Air movement acceptability limits and thermal comfort in Brazil's hot humid climate zone. Building and Environment, doi:10.1016/j.buildenv.2009.06.005.

Carrier Corporation Software Systems Network (1998). HAP quick reference guide.

Causone F., Baldin F., Olesen B.W., Corgnati S.P. (2010). Floor heating and cooling combined with displacement ventilation: possibilities and limitations. Energy and Buildings, 42, 2338-2252.

Chen, Q. (1995). Comparison of Different k-ξ Models for Indoor Airflow Computations: Numerical heat Transfer, 28(Part B), 353-369.

Chow, T.T., Chan, A. L. S., Fong,K.F., Lin,Z., He, W., Ji, J. (2009) “Annual performance of building-integrated photovoltaic/water-heating system for warm climate application,” Applied Energy, 86 (5), 689–696.

Chow, T.T., Fong, K.F., Givoni, G., Lin, Z. and Chan, A.L.S. (2010). Thermal Sensation of Hong Kong people with increased air speed, temperature and humidity in air-conditioned environment. Building and Environment, 45, 2177–2183.

Chow, T.T., Ji, J. (2012). Environmental Life-Cycle Analysis of Hybrid Solar Photovoltaic/Thermal Systems for Use in Hong Kong,” International Journal of Photoenergy, Volume 2012, Article ID 101968, 9 pages, doi:10.1155/2012/101968

Chung, T.M., and Tong, W.C. (1990) Thermal Comfort Study of Young Chinese People in Hong Kong. Building and Environment, 25, 317–328.

Cohen, J.B., Allan, J.R. and Sowood, P.J. (1989) Effect of head or neck cooling used with a liquid-conditioned vest during simulated aircraft sorties, Aviation Space Environmental Medicine, 60, 315–320.

Cornish, A. et al. (2011). Hong Kong Ecological Footprint Report 2010. World Wild Fund, HKSAR.

Davies, M.G. (1993). Definitions of room temperature, Build Environ, 28, 383–398.

Defra (2008). PAS 2050 - Specification for the Assessment of the Life Cycle Greenhouse Gas Emissions of Goods and Service. Department for Environment, Food and Rural Affairs, UK.

Definition of Life cycle assessment website. Retrieved June 2014, from http://en.wikipedia.org/wiki/Life-cycle_assessment

“eBalance” handbook for Life Cycle Cost Analysis, 亿科环境科技有限公司（IKE）, China. (2014 Chinese version). Retrieved October 2014, from http://www.itke.com.cn/help/ebalance/

Ecoinvent Database (2014). Retrieved October 2014, from http://www.ecoinvent.org/database/.

Electrical and Mechanical Services Department - Energy Analysis and Saving Technologies: Life Cycle Energy Analysis, LCEA of Building Construction (2006). Retrieved January 2014, from http://www.emsd.gov.hk/emsd/eng/pee/lceabc.shtml

Electrical and Mechanical Services Department - Energy Conservation Charter 2006- Suitable Room Temperature. Retrieved October 2008 from http://www.energyland.emsd.gov.hk/blue_sky/eng/index.htm.

Electrical and Mechanical Services Department - Promote the application of Building Energy Code. Retrieved June 2013 from http://www.beeo.emsd.gov.hk/en/pee/BEC_2012.pdf

Electrical and Mechanical Services Department - HKSAR Energy Efficiency Registration Scheme for Buildings. Retrieved September 2014 from http://www.emsd.gov.hk/emsd/eng/pee/eersb.shtml

ELCD- European reference Life Cycle Database (2014). Retrieved October 2014, from http://eplca.jrc.ec.europa.eu/ELCD3/

Elvsen, P. and Sandberg, M.(2009). Buoyant jet in a ventilated room: Velocity field, temperature field and airflow patterns analysed with three different whole-field methods. Building and Environment 44, 137-145.

Envest 2 – Environmental impact assessment & whole life cost tool. Retreived from August 2014 from http://envestv2.bre.co.uk/account.jsp

Fanger P.O., Christensen N.K. (1986). Chart for prediction of draught. Australian Refrigeration, Air Conditioning and Heating, 40(7): 46.

Fanger, P.O. (1972). Thermal Comfort, NT. USA, McGraw-Hill.

Fluent on-line User's Guide (2013). Retrieved October 2011 from: http://jullio.pe.kr/fluent6.1/help/html/ug/node1.htm.

Fong, K.F., Chow, T.T., Li, C. (2010). Comfort zone of air speeds and temperatures for air-conditioned environment in the subtropical Hong Kong. Indoor and Built Environment, 19(3), 375-381.

Fong, K.F., Chow, T.T., Li, C., Lin, Z., Chan, L.S. (2010). Effect of neutral temperature on energy saving potential of centralized air-conditioning systems in subtropical Hong Kong. Applied Thermal Engineering, 30(13), 1659-1665.

Fong, K.F., Yuen, S.Y., Chow, C.K., Leung, S.W. (2010). Energy management and design of centralized air-conditioning systems through the non-revisiting strategy for heuristic optimization methods. Applied Energy, 87(11), 3494-3506.

Fong M.L., Lin Z., Fong, K.F., Chow, T.T., Yao T. (2011). Evaluation of thermal comfort conditions in a classroom with three ventilation methods. Indoor Air 2011; 21(3): 231-239.

Fong, M.L.(2012). Environmental Protection by elevated indoor temperature of air conditioning system. The 2012 international symposium on public health and environmental protection, May 2012.

Fong, M.L., Fong, K.F., Lin, Z. (2010). Experimental study of determining neutral temperatures for conventional mixing and stratum ventilation modes in environmental chamber. The 5th Asian Conference on Refrigeration and Air-conditioning, June 2010.

Forster, P. et al. (2007), Changes in Atmospheric Constituents and in Radiative Forcing. In: Climate Change 2007: The Physical Science Basis. Contribution of Working Group I to the Fourth Assessment Report of the Intergovernmental Panel on Climate Change. Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA.

Fountain ME, Arens EA, de Dear R, Bauman F, Miura K. (1994) Locally controlled air movement preference in warm isothermal environments. ASHRAE Transactions; 100:937–52.

Fountain ME, Arens EA.(1993). Air movement and thermal comfort. ASHRAE Journal August 1993, 26-30.

Geoff Hammond and Craig Jones (2011). Inventory of carbon& energy, ICE version 2.0: Summary Tables. Retrieved January 2014 from http://web.mit.edu/2.813/www/readings/ICEv2.pdf.old

Givoni, B. (2005). Responses to humidity in hot humid climates, Proceedings of the 10th international conference on indoor air quality and climate, Indoor Air 2005, Beijing. (CD ROM)

Givoni, B. (1998). Climate considerations in Building and Urban Design. NY, Van Nostrand Reinhold.

Fountain ME, Arens EA, de Dear R, Bauman F, Miura K. (1994). Locally controlled air movement preference in warm isothermal environments.
Hammond, G. P. and Jones, C. I. (2008), Embodied energy and carbon in construction materials. Proceeding Institution of Civil Engineers: *Energ y*. 161(2), 87-98.

Haapio, A and Viitaniemi, P. (2008). A critical review of building environmental assessment tools. Environmental Impact Assessment Review, 28, 469–482

HKSAR Government, Life Cycle Assessment, LCA and Life Cycle Costing, LCC Tool for Commercial Building Developments in HKSAR: User Manual (2005). EMSD, HKSAR SAR Government publication. Retrieved August 2013 from http://www.emsd.gov.hk/emsd/eng/pee/lceabc.shtml

Guidelines to Account for and Report on Greenhouse Gas Emission and Removals for Buildings (Commercial, Residential or Institutional Purposes) in HKSAR, EMSD and EPD, HKSAR Government publication, 2010.

HKSAR Government - Buildings Energy Efficiency Ordnance. Retrieved September 2012 from http://www.legislation.gov.hk/blis_ind.nsf/WebView?OpenAgent&vwpg=CurAllEngDoc*607*100*607.1#607.1

Hui, Z., et.al. (2007). Air movement preferences observed in office buildings, International Journal Biometeorol, 51, 349-360.

International Facility Management Association. Temperature wars - savings vs comfort. Retrieved July 2010 from http://www.ifma.org.

International Organization for Standardization (2006). ISO 14040: Environmental Management Life Cycle Assessment - Principles and Framework. Switzerland.

International Organization for Standardization (2006). ISO 14044: Environmental Management - Life Cycle Assessment - Requirements and Guidelines. Switzerland.

International Panel on Climate Change (2006). Guidelines for National Greenhouse Gas Inventories. Japan.

ISO 7730 (1995). Moderate thermal environments-determination of the PMV and PPD indices and specification of the conditions for thermal comfort. Geneva: International Standards Organization.

ISO 14000 website information. Retrieved June 2012 from, http://en.wikipedia.org/wiki/ISO_14000

Jackman, P.J. (1990). Displacement ventilation, BSRIA, Technical Memorandum 2/90.

Japanese Ministry of the Environment- National campaign to fight global warming. Retrieved September 2010 from http://www.env.go.jp/en/press/2005/0428b.html.

Kalogirou, S.A. (2009). Thermal performance, economic and environmental life cycle analysis of thermosiphon solar water heaters. Solar Energy,83,39-48

Kam, M.S. (2009). 25.5 deg c and human comfort. Retrieved October 2013, from http://www.emsd.gov.hk/emsd/e_download/wnew/conf_papers/HKIE_Environment_Annual_Seminar_Paper_25.5_deg_C_and_Human_Comfort.pdf.

Karimipanah, T., Awbi, H.B., Moshfegh, B. (2008). The air distribution index as an indicator for energy consumption and performance of ventilation system. Journal of the Human-Environment System, 11(2), 77-84.

Kelly, K.A., McManus, M.C., Hammond, G.P. (2012). An energy and carbon life cycle assessment of tidal power case study: The proposed CardiffeWeston severn barrage scheme, Energy, doi:10.1016/j.energy.2012.05.023

Kline, S.J., MCclintock, F.A. (1953). Describing uncertainties in single-sample experiments. Mechanical Engineering, 75: 3-9.

Koestel, A., Tuve, G.L. (1955). Performance and evaluation of room air distribution systems. ASHRAE Transactions, 61: 533.

Kosonen, R. (2010). Displacement ventilation for room air moisture control in hot and humid climate air distribution techniques (Air distribution techniques 5), HVAC handbook —Displacement Ventilation in Hot and Humid Climate.

Launder, B.E. and Spalding,D.B. (1974). The Numerical Computation of Turbulent Flows” Comp. Meth. Appl. Mech., 3, 269-289

Li, L. (2005). The improvement of summer thermal environment of ventilated-room of Xiamen in China. Proceedings of the 10th international conference on indoor air quality and climate; 199–203.

Lin, Z., Wang, J.L., Yao, T., Chow, T.T. (2012). Investigation into anti-airborne infection performance of stratum ventilation. Building and Environment; 54, 29-38.

Lin, Z. (2011). Effective draft temperature for evaluating the performance of stratum ventilation. Building and Environment, 46, 1843-1850

Lin, Z., Lee, C.K., Fong, K.F., Chow, T.T. (2011). Comparison of annual energy performances with different ventilation methods for temperature and humidity control. Energy and Buildings, 43: 3599-608.

Lin, Z., Lee, C.K., Fong, K.F., Chow, T.T., Yao, T., Chan, L.S.A. (2011). Comparison of annual energy performances with different ventilation methods for cooling. Energy and Buildings, 43: 130-136.

Lin, Z., Yao, T., Chow, T.T., Fong, K.F., Chan, L.S.A. (2011). Performance evaluation and design guidelines for stratum ventilation. Building and Environment, 46: 2267-2279.

Lin, Z., Tian, L., Yao, T., Wang, Q., Chow, T.T. (2011). Experimental and numerical study of room airflow under stratum ventilation. Building and Environment, 46, 235-244.

Lin, Z., Chow, T.T., Tsang, C.F., Fong, K.F., Chan, L.S. (2010). Stratum ventilation for a workshop under elevated indoor temperature. International Journal of Ventilation, 9(1): 47-58.

Lin, Z., Chow, T.T., Tsang, C.F., Fong, K.F. and Chan, L.S. (2009). Stratum ventilation a potential solution to elevated indoor temperature,Build Environ,44, 2256–2269.

Lin, Z., Chow, T.T., Tsang, C.F., Fong, K.F., Chan, L.S.A. (2005). CFD study on effect of the air supply location on the performance of the displacement ventilation system, Building and Environment, 40(8): 1051-1067.

Lin, Z., Chow, T.T. and Tsang, C.F. (2005). Stratum ventilation - A conceptual introduction. Proceedings of the 10th International Conference on Indoor Air Quality and Climate, Beijing, China, 4–9 September 2005, 3260–3264.

Mayer, E. (1992). New measurements of the convective heat transfer coefficients: influences of turbulence, mean air velocity and geometry of human body. Proceedings of the ROOMVENT 92, vol.3, Lyngby, Danish Association of HVAC Engineers (DANVAK), 263–276.

Mayer, E. and Schwab, R. (1988). Direction of low turbulent airflow and thermal comfort. Proceedings of the Healthy Buildings 88, Vol. 2, Stockholm, Sweden,577–582.

Melikov, A.K. and Nielsen, J.B. (1989). Local thermal discomfort due to draught and vertical temperature difference in rooms with displacement ventilation. J.ASHRAE Transactions, Vol. 95, Part. 2, VA-89-19-5.

Memarzadeh, F,, Manning, A. (2000). Thermal comfort, uniformity, and ventilation effectiveness in patient rooms: performance assessment using ventilation indices. ASHRAE Transactions 2000; 106(2): 748-61.

Miller, P.L.(1989). Descriptive methods in Building systems: Room air and air contaminant distribution. L.L. Christianson ed. ASHRAE.

Ministry of Knowledge and Economy, the Republic of Korea (2008). The standard of energy management. Retrieved December 29, 2008 from http://www.mke.go.kr/info/law/gosiView.jsp? (Item No. 2008-219).

Mui, K.W. and Wong, L.T. (2007). Neutral temperature in subtropical climates – a field survey in air-conditioned offices, Build Environ, 42, 699–706.

Nakamura, M., Yoda, T., Crawshaw, L.I., Yasuhara, S., Saito, Y., Kasugam M. (2008). Regional differences in temperature sensation and thermal comfort in humans. Journal of Applied Physiology; 105(6): 1897-1906.

Nicol, F. (2004). Building research and the sustainability of the built environment in the tropics. Energy and Buildings, 36(7):628–37.

National Development and Reform Commission (NDRC 2005). The State Council of China, "Citizen's Pact for Deed of Thrift". Retrieved July 2008 from http://www.ndrc.gov.cn/gzdt/t20051223_54462.htm.

NOW news, Taiwan, Campaign in office of president for saving energy and reducing carbon emissions. Retrieved June 2008 from http://www.now-news.com/2008/05/24/10844 -227 9584.htm.

Nunneley, S.A., Reader, D.C. and Maldonado, R.J. (1982). Head-temperature effects on physiology, comfort, and performance during hyperthermia, Aviat. Space Environ. Med., 53, 623–628.

Olesen, B.W., Koganei, M., Holbrook, G.T., Woods, J.E. (1994). Evaluation of a vertical displacement ventilation system. Building and Environment, 29(3): 303-10.

Peng, J.Q., Lin. L., Yang, H.X. (2013). Review on the life cycle assessment of energy payback and greenhouse gas emission of solar photovoltaic system. Renewable and Sustainable Energy Review 2013,19:255-274.

Rydberg, J, Norback, P. (1949). Air distribution and draft. ASHRAE Transactions 1949; 55: 225.

Salim, M. and Cheah, S.C. (2009). Wall y+ strategy for dealing with wall-bounded turbulent flows. Proceedings of the International Multiconference of Engineers and Computer Scientists 2009 Vol II, March 2009, HKSAR.
Srivajana, W. (2003). Effects of air velocity on thermal comfort in hot and humid climates, Thammasat Int J Sci Tech, 8, 2.

Straub, H.E., Chen, M.M. (1957). Distribution of air within a room for year-round air conditioning, Part Ⅱ. University of Illinois Engineering Experiment Station Bulletin, 442; 1957.

Straub, H.E., Gilman, S.F., Konzo, S.(1956). Distribution of air within a room for year-round air conditioning, Part Ⅰ. University of Illinois Engineering Experiment Station Bulletin, 435; 1956.

Schild, P.G. (1997). Accurate Prediction of Indoor Climate in Glazed Enclosures, Ph.D. Thesis. Norwegian University of Science and Technology (NTNU), Norway.

SimaPro - Reliable Databases and Methods. Retrieved July 2014 from http://www.pre-sustainability.com/databases

Tanabe, S., Kimura, K.(1989). Thermal comfort requirements under hot and humid conditions. In: Proceedings of the first ASHRAE Far East conference on air conditioning in hot climates. Singapore: ASHRAE; 1989. p. 3–21.

Tian, L., Lin, Z., Liu, J. and Wang, Q.W. (2008). Numerical study of indoor air quality and thermal comfort under stratum ventilation, Prog. Comut. Fluid Dyn., 8, 541–548.

Tian, L. (2009). Experimental and numerical study of indoor environment under stratum ventilation. Doctoral Thesis. Xi'an Jiaotong University.

Tian L., Lin, Z., Wang, Q.W. (2011). Experimental investigation of thermal and ventilation performances of stratum ventilation. Building and Environment, 46, 1309–1320.

Tian, L., Lin, Z., Liu,J., Yao. T., Wang, Q.W.(2010). The impact of temperature on mean local air age and thermal comfort in a stratum ventilated office, doi:10.1016/j.buildenv.2010.08.013.

Tian, L., Lin, Z., Wang, Q.W., Liu, J.(2010). Comparison of gaseous contaminant diffusion under stratum ventilation and under displacement ventilation, Building and Environment, 45 (9), pp. 2035–2046
Tian, L., Lin, Z., Wang, Q., Liu, J. (2009). Numerical investigation of indoor aerosol particle dispersion under stratum ventilation and under displacement ventilation. Indoor and Built Environment, 18(4): 360-375.

Toftum, J. (1997). Effect of airflow direction on human perception of draught. Proceedings of CLIMA 2000, p. 11.

Wang, Q.W., Zhao, Z. (2006). Performance comparison between mixing ventilation and displacement ventilation with and without cooled ceiling. Engineering Computations, Vol. 23, No.3, pp. 218-237.

Williams, B.A. and Chambers, A.B. (1971). Effect of neck warming and cooling on thermal comfort, NASA Special Publications SP-302: 289–294.

WRI and WBCSD (2011). Greenhouse Gas Protocol - Product Life Cycle Accounting and Reporting Standard. World Resources Institute and World Business Council for Sustainable Development

Wu, Y.L., Hsu, S.P., Sung, W.P., Hsieh, W.D. (2010). Effect of diffuser design on thermal comfort and ventilation efficiency. Practice Periodical of Hazardous, Toxic, and Radioactive Waste Management, 14(4): 229-33.

Wyon, D.P., Sandberg, M. (1990). Thermal manikin prediction of discomfort due to displacement ventilation. ASHRAE Transactions 1990: 96 (1)

Yakhot, V. and Orszag, S.A. (1986). Renormalization Group Analysis of Turbulence. I. Basic Theory. Journal of Scientific Computing, boulume 1, 1, 3-51.

Yamtraipat, N., Khedari, J., Hirunlabh, J. (2005). Thermal comfort standards for air conditioned buildings in hot and humid Thailand considering additional factors of acclimatization and education level. Solar Energy, 78:504-17

Yoshihide, T., Stathopoulos,T. (2007). Turbulent Schmidt numbers for CFD analysis with various types of flowfield, Atmospheric Environmental, 41, 8091-8099.

Yuan, X., Chen, Q., Glicksman, L.R., Hu, Y., Yangm X. (1999). Measurements and computations of room airflow with displacement ventilation. ASHRAE Transactions 1999; 105(1): 340-352.

Yu, W.J., Cheong, K.W.D., Kosonen, R., Xie, Y.H., Leow, H.C. (2003). A thermal comfort study on displacement ventilation in the tropics. Proceedings of Healthy Buildings, 7th International Conference, Singapore 2003; 2: 42-7.

Zhang, H., Arens, E., Fard, S.A., Huizenga, C., Paliaga, G., Brager, G. (2007). Air movement preferences observed in office buildings. International Journal of Biometeorology, 51(5):349-60.

Zhang, H., Arens, E., Kim, D.E., Buchberger, E., Bauman, F., Huizenga, C. (2008). Comfort, perceived air quality, and work performance in a low-power task-ambient conditioning system. In: International symposium on the interaction between human and building environment. July 2–3, Yonsei University, Korea; 2008.

APPENDIX A: CONSTRUCTION, TESTING AND COMMISSIONING OF ENVIRONMENTAL CHAMBER

A1 Sustainable system construction

Completing of design, tender & procurement stage, the following construction works together with Testing and commissioning process were successfully completed under awarded funding of HK$1,850,000 in 2008. Figure A1 shows the cost breakdown of the development of intelligent Project Studio in BST design studio at my local university for supporting my study. All assigned contractors and specified materials/ devices/equipment installed inside the environmental chamber were implemented in line with the milestones of this research.

The Development of an Intelligent Project Studio — *Page* 11

6. Cost Estimation

Budget Cost – HK$ 1,850,000

Checked by EO | BST

a) HK$ 697,450 for building services installation as mentioned in Section 5.1 above. Please refer to the quotation by FMO dated 11 August 2008 in Appendix VI.

b) HK$ 702,425 for building management system as mentioned Section 5.2 above. Please refer to the quotation from Hensen System Engineering Limited dated 15 August 2008 in Appendix VII.

c) HK$ 188,676 for Power Quality and Energy Monitoring System as mentioned Section 5.2 above. Please refer to the quotation from Powerpeg NSI limited dated 30 October 2008 in Appendix VIII.

d) HK$ 135,000 for BACnet integration for exiting BMS system as mentioned Section 5.2 above. Please refer to the quotation from Johnson Controls dated 14 November 2008 in Appendix IX.

e) HK$ 31,890 for lighting fitting as mentioned Section 5.2 above. Please refer to the quotation from GELEC(HK) Limited dated 18 Nov 2008 in Appendix X.

f) HK$ 94,559 as provisional sum for history server, web server, operation workstation and wireless router, UMPC & PDA devices as shown in Conceptual Architecture of the Building Management System under Appendix IV.

Prepared by	Endorsed by	Approved by
Ir. Alan Fong	Dr. Norman Tse	Dr Paul Ho
Instructor I	Programme Leader, AScBSE	Head, BST
Date: 19 NOV 2008	Date: 19 Nov. 2008	Date: 20/11/2008

Approved by

Prof C H Chan
Dean (FSE)

Date: Dec 9, 2008

Figure A1 Approval letter of environmental chamber development

A1.1 BMS with monitor and control device

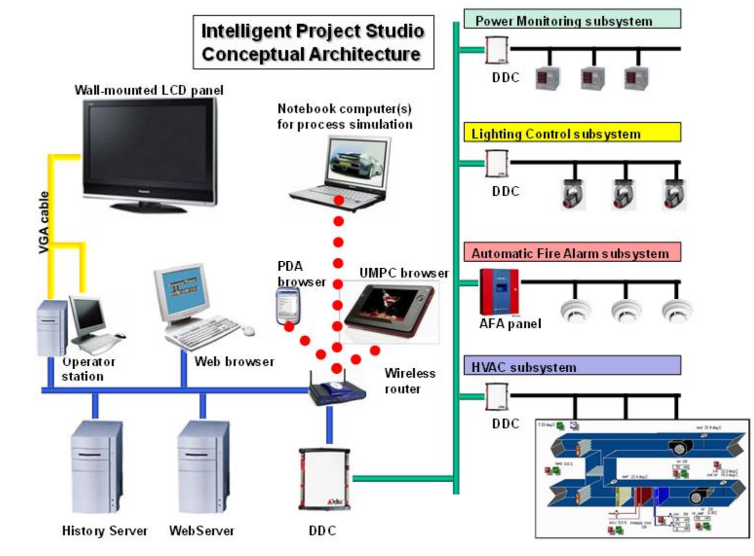


Figure A2 Conceptual architecture of BMS

The conceptual architecture of BMS is shown in Figure A2 and its features are listed out as follows:

1. Set up workstations with web server, historian server, IE Browser, wireless and wired access control and monitoring device, minimum 42’ LED display panel. Wireless applications can be effected by UMPC.
2. Install all interfacing devices such panels / dry contacts for monitoring CityU main building services systems, such as main chiller plants and lift & escalator systems.
3. Standardize the control and communication means for various systems, by using BACnet protocol, MODBUS and RS232.
4. Supply all the control and measuring devices such as energy meter, air temperature sensor, RH sensor and other devices.

A1.2 Power Quality Monitoring System

It is used to monitor the following four main parameters via seven smart meters as shown in Figure A3.

1. Real time power information: rms values of phase & line voltages and currents, displacement and total power factors and the leakage currents.
2. Energy data: kWh, maximum kW, kVA and kVAr with time stamps
3. Event data: minimum/maximum values with time stamp up to 1 second record for the above items (i) and (ii) above, sag/swell voltage or setpoint current trigger up to half cycle calculation and transient events. On/off/trip and busbar temperature +monitoring are also included in the critical event monitor points as well.
4. Power harmonics: THD of voltage and current harmonics; individual odd current harmonic measurement may extend to the 21st and neutral to earth voltage.

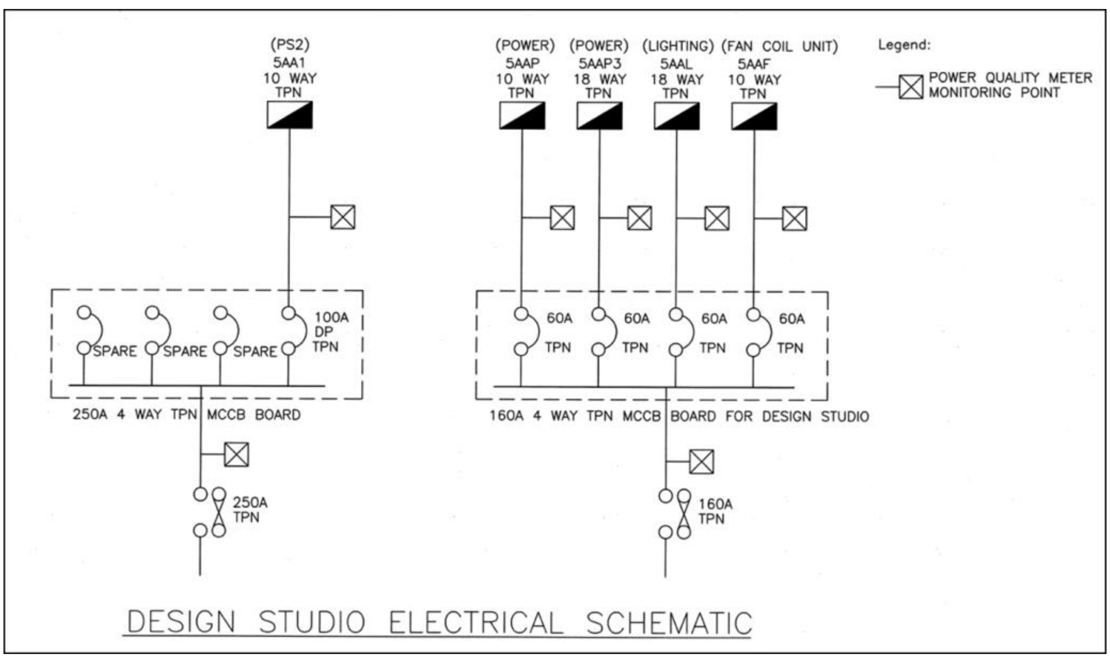


Figure A3 Power quality monitoring system

A1.3 Ventilation systems

Mixing, displacement and stratum ventilation system are installed in this environmental chamber with all monitoring and control parameter as shown in Figure A4 and A5.

1. Ceiling mounted AHU with variable speed of supply fan.
2. CO_2 sensor for fresh air control.
3. Air distribution device such as air ductwork, insulation, VAV box and air diffusers, etc.
4. Acoustics treatment.
5. Water distribution device such as chilled water pipe, insulation, control valve, etc.
6. Monitoring & control device for assessing thermal comfort and energy consumption.

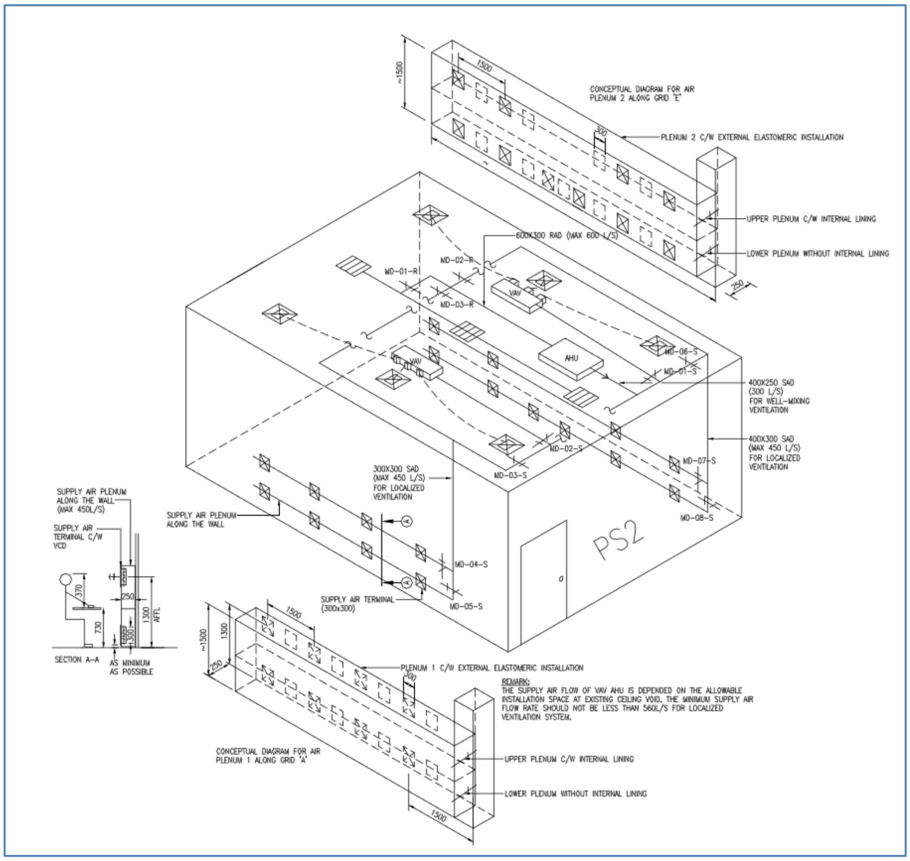


Figure A4 Construction concept of enviormental chamber

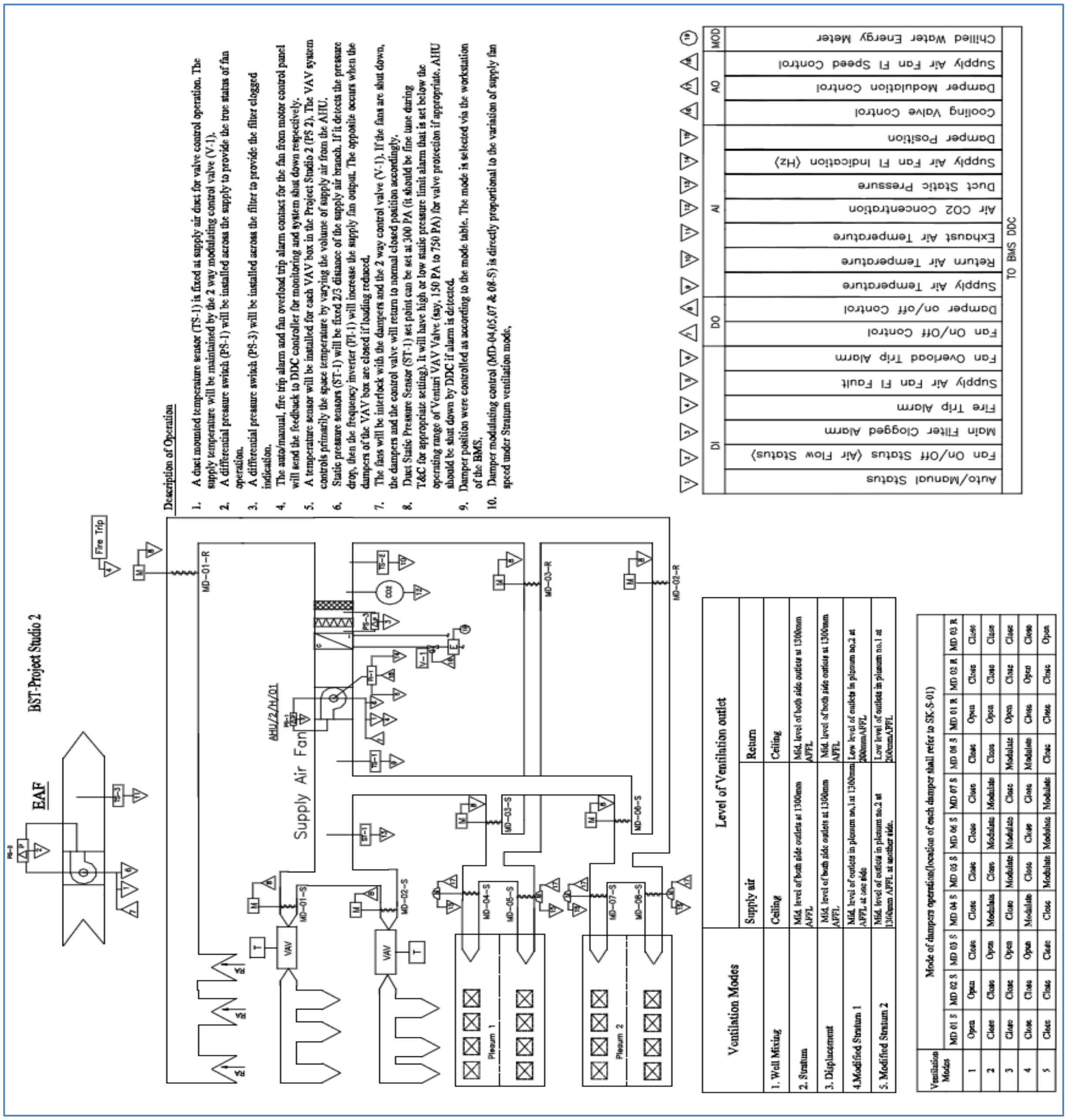


Figure A5 Point schedule and mode table for mixing, displacment and stratum ventilation.

A2 Testing and commissioning

In order to ensure good performance of each equipment/device/component of various air distribution system to be successfully commissioned inside environmental chamber device, all technical requirements of the commissioning works has been defined clearly and has been provided with adequate information, documented in the form of drawings, schedules and specifications, for instance:

1. The scope of the works, i.e. the systems to be commissioned, their design philosophy, functions and duration of operation, and an explanation of their operational inter-relationship with other engineering services systems.
2. The setting out of the responsibilities of the various parties (client, design team, main or managing contractor, installation contractor and commissioning specialist). BSRIA Technical Memorandum 1/88, "Commissioning HVAC Systems - Division of Responsibilities" gives useful guidance.
3. The technical specification of the commissioning work. For example: the standards (e.g. relevant parts of CIBSE Codes and BSRIA Guides) to which the works being carried out;
    - *i.* the diversity pattern to be applied during commissioning;
    - ii. the extent of factory calibration of each components;
    - iii. the extent of site testing of installed components;
    - iv. the monitoring and control facilities to be provided by the BMS;
    - v. the tolerances for regulation and for test results;
    - vi. the reporting procedures required.

4. The witnessing procedures to be observed. These has been specified whether the witnessing has been undertaken during the commissioning process or whether the results are to be demonstrated after commissioning is complete.
5. Design drawings showing the layout of the system in relation to the building form, and if required, other engineering services.
6. Schematic diagrams clearly illustrated the design intent and including all the design information required to commission the system. The reference identification is unique to each individual component to permit cross referencing and to enable a component to be identified in correspondence and telephone discussions. Schematics prepared with the design drawings can reveal potential difficulties in commissioning and enable these to be rectified prior to installation.
7. Schedules of major plant, equipment and components, cross referenced to the design drawings and schematic diagrams.
8. Additional design information required for commissioning, which may not be available until after the appointment of the building services installer, for example:
    i. electrical wiring diagrams of associated plant and equipment;
    ii. operational details of each components;
    iii. control system diagrams;
    iv. fan duty control systems and fan performance curves;
    v. working drawings (see Figure A6).

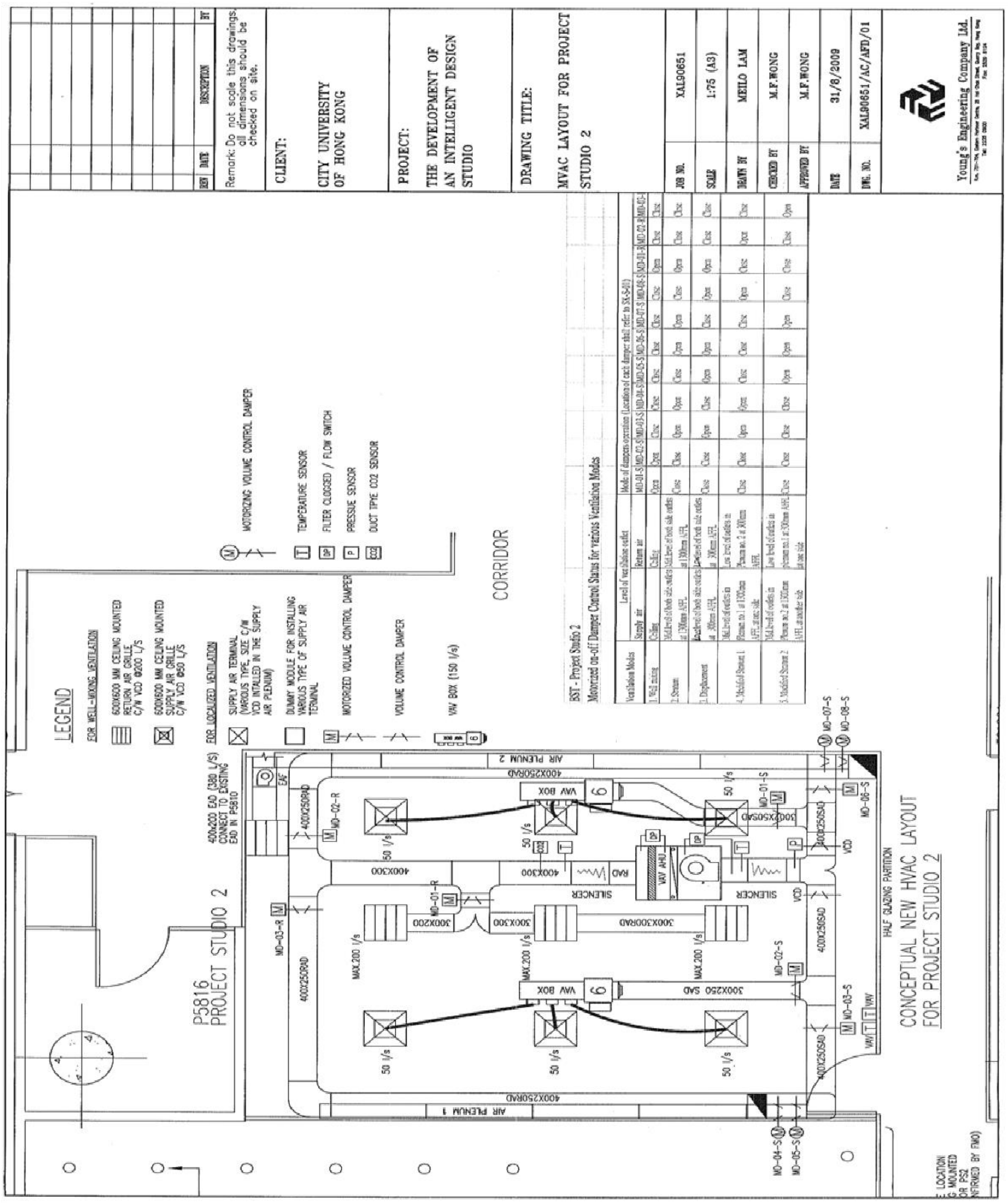


Figure A6 Working drawing for the environmental chamber

A2.1 Preliminary Procedure for air distribution System Balancing

1. Mechanical Work

    i. to check system cleanliness
    ii. to ensure the related air regulating devices in open position and free to move
    iii. to check for air tightness
    iv. to ensure the related fan impellers are secure and free to move
    v. to ensure the pulley and coupling with secured and alignment being adjusted
    vi. to satisfy the belt tension
    vii. to drain water from the drain pan to check the drainage function
    viii. to check the internal fan vibration isolator is in free suspended status

2. Electrical Work


Photo A1: Control panel


Photo A2: Fuse rating

    i. to ensure the local isolation provision for motor and each control circuit / MCB
    ii. to ensure the cleanliness of the panels and switches gear (see Photo A1)
    iii. to ensure corrective the fuses rating provided (see Photo A2)
    vii. to ensure corrective starter overloads matched with the motor nameplate full load current
    viii. to satisfy the insulation test on motor
    ix. to test the control circuit logic and starter operation

x. to check the emergency stop function
xi. to check the fire alarm interlock function

3. Setting to Work

i. to check that the direction and speed of rotation of motor shaft are correct
ii. to check that the motor running current and voltage on all phases
iii. to check and record IN/OUT temperature of water side
iv. to check and record that the IN/OUT pressure of waterside and to determine the water flow through the cooling coil.
v. to check and record the supply / return air temperature
vi. to check and record the fan speed, static and airflow against the design.
vii. to check the emergency stop for fan motor, overload setting and testing
viii. to check interlock cutout i.e. fire trip, and volume control damper actuation.

4. System cleanliness of system components including

i. air intake screens
ii. Fan and other equipment chamber
iii. Fan internals
iv. Cooling coil trays
v. Washer tanks
vi. Ducting and other airways (see Photo A3)

Photo A3: Air ducting under construction

5. Sensing elements

    i. to ensure all sensing elements installation (see photo A4 to A11) matched with the specification and predicated design performance
    ii. to ensure the installed location and methodology in compliance with the manufacturer recommendation and design intent

Photo A4: Airflow sensor

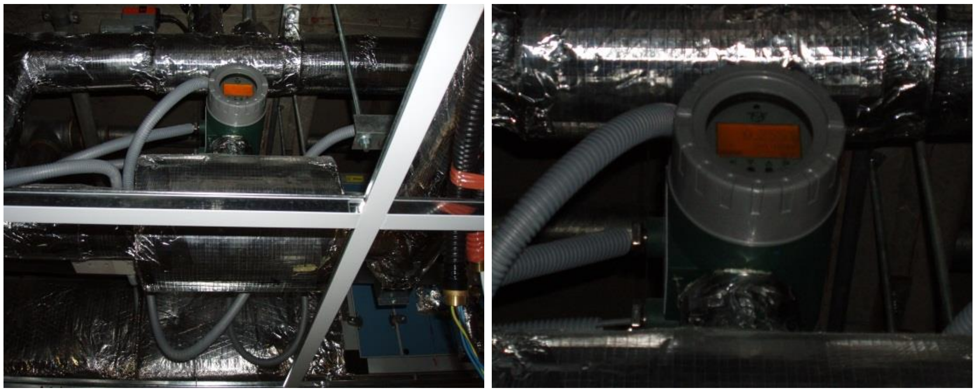


Photo A5: Energy meter installed in chilled water pipe

Photo A6: RH sensor

Photo A7: Temperature sensor

Photo A8: Wall mounted thermostat

Photo A9: Room air pressure sensor

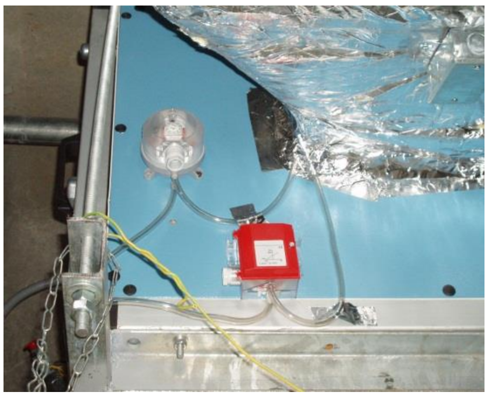


Photo A10: Air pressure sensor

Photo A11: Water pressure gauge

5. VAV Box

i. to ensure all VAV box installation (see photo A12 to A15) matched with the specification and predicated design performance

ii. to ensure the installed location and methodology in compliance with the manufacturer recommendation and design intent

Photo A12: VAV box outlook

Photo A13: Actuator of VAV box

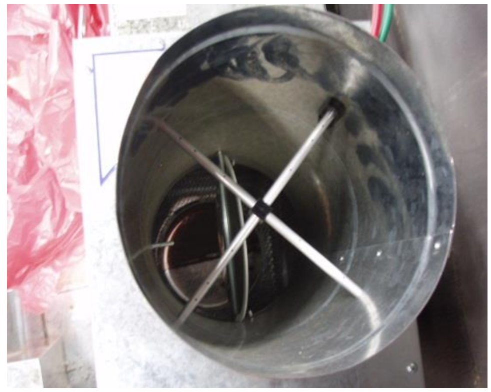


Photo A14: Outlet of VAV box

Photo A15: VAV box location and wiring construction

7. Checking of all material/devices/equipment installed in air conditioning system

   i. to ensure all turning vanes, thermal insulation, acoustic linings, battery fins and sensing elements to be fitted and undamaged.

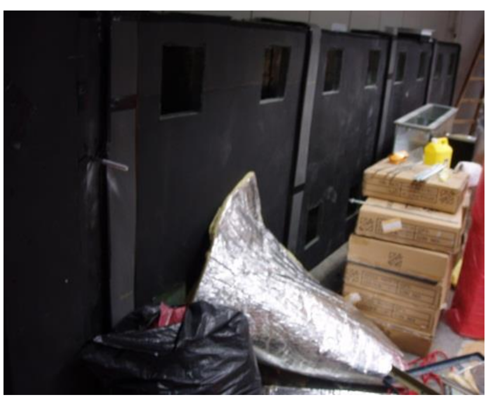


Photo A16: wall mounted air plenum

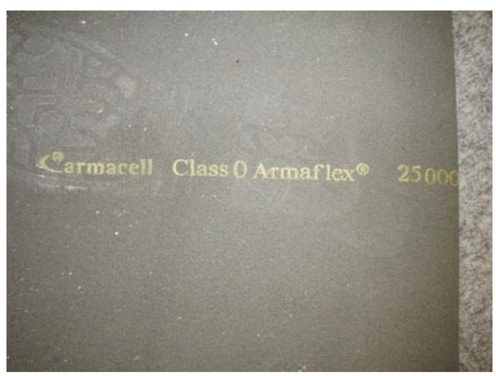


Photo A17: “Armaflex” insulation

Photo A18: Instilling of ductwork components

Photo A19: Acoustic lining inside wall mounted air plenum

ii. Filters, silencers, dampers, etc., installed correctly in relation to the direction of air flow (see Photo A20 & A21)

Photo A20: Filter at AHU

Photo A21: Ductwork silencer

iii. Damper free-movement, clearances, seating, pinning to damper spindles, position of blades with respect to quadrant indication, relative positions of blades in multi-leaf dampers (see Photo A22).


Photo A22: Electric actuator for air damper control

iv. Free movement of fire dampers if exist together with the location of access to and fitting of fusible link assembly all fire dampers finally secured in open position.

v. Dampers throughout the system secured in suitable position of different mode condition (see Photo A23).

Photo A23: Fire damper

vi. All adjustable louvers set without deflection, i.e. normal to face of grille. Adjustable cones on diffusers set either all in the fully up or all in the fully down position (see Photo A24).

Photo A24: Diffuser

vii. Test holes provided for measurement of total airflow; cleansing air and testing point at ductwork Cleansing and testing point at air plenum (see Photo A25)

Photo A25: Test pints along all ductwork

8. Builder’s work and shafts sealed

   i. To ensure all builder’s work done properly (see Photo A26)

Photo A26: Builder’s work

9. Dummy air inlets/outlets sealed

   i. To ensure the tightness of all dummy (see Photo A27)

Photo A27: Dummy sealed

10. Ductwork joints, including flexible couplings

    i. To ensure all joints' installed properly (see Photo A28)

Photo A28: Flexible ductwork

11. Inspection covers fitted and plugs or covers for test holes fitted (see Photo A29)

Photo A29: Cover for test hole

A2.2 Procedure for Air Distribution System Balancing

1. Determine that all preliminary procedures have been preformed
2. Determine that all air circuits are functional
3. Place all related supply, exhaust and return air systems in operation with the fans running at design RPM
4. Establish system conditions for the maximum demand in airflow.
5. Make sure that both supply and exhaust systems serving each space are operating

6. Determine (or set up so) that automatically controlled devices (see Photo A30) in the system will not adversely affected the balancing operations

Photo A30: Damper controller

7. Determine that volume flow rate of air using Pilot tube (see Photo A31) transverse of main duct

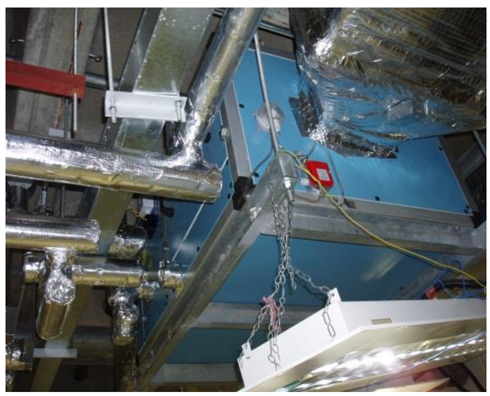


Photo A31: Pilot tube

8. Or using Fan curve of fan performance charts. In order to determine fan performance using a fan curve or performance rating chart, it is necessary to take amperage and voltage readings and calculate brake horsepower. In addition, a static pressure reading across the fan must be recorded. With RPM, brake horsepower and static pressure, the fan manufacturer's data sheets may be used to determine the air flow rate predicted by manufacturer.
9. Anemometer readings across coils or dampers on the intake side of the fan. This is used as an approximation
10. Summation of the air quantity at all outlets/ inlets for different ventilation modes.

11. If the fan volume is not within plus or minus 10% of design capacity at design RPM, determine the reason by reviewing all system conditions, procedures and recorded data. Check and record the air pressure drop across filter coils, sound traps etc., to see if excessive loss is occurring. Particularly study duct and casing conditions of the fan's inlet and outlet. If large discrepancy is found between the measured air volume and design flow rate, carry out duct static loss calculation according to the installed duct route to verify the required RPM by frequency inverter (see Photo A32) for field balancing.

Photo A32: Frequency inverter

12. Make Pitot tube traverses to all-main supply ducts test hole to determine air distribution. Investigate any branch that is very low in capacity to make sure that no blockage exists. Ensure that all VCD or fire dampers should be opened before commencement of work. All adjustable louvers should be set without deflection. Check and record the readings (see Photo A33 to A35).

Photo A33: Electric type VCD

Photo A34: Manual type VCD

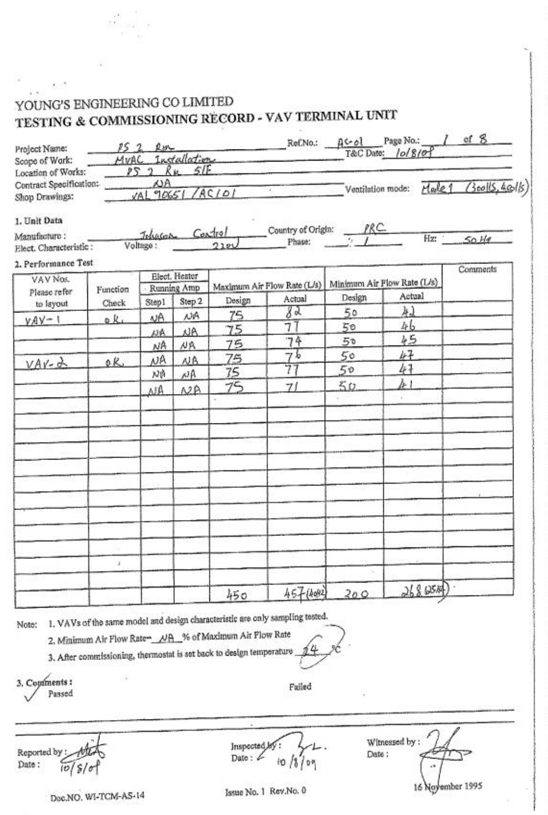
Photo A35: Measurement

13. Adjust the volume dampers in the main ducts to the appropriate air flow requirement
14. If the measured airflow is not the designed flow rate, adjust the diffuser or grille until airflow reaches the design flow rate. Plan the sequence of branch (diffuser/grille) balancing. In making the adjustments it is preferable to adjust the scoop devices rather than the dampers at the air terminals (see Photo A36).

Photo A36: Air flow measurement by hood rotating vane anemometer

15. The branch which has the highest indicated percentage of measured airflows will be tackled first. The sub-branch with the highest percentage of measured airflow that tee-off from the branch will be regulated first.
16. Regulation will be started at the least favored terminal and its percentage of design airflow is used to proportional-balance the group of terminals along this sub-branch.
17. The sub-branch with the second highest percentage of measured airflow will be regulated as mentioned in item xii above and so on.

18. When the terminals at all sub-branches have been balanced, regulation for this group of sub-branch is preceded.
19. Procedures xiii to xv are repeated for the branch with the second highest percentage of measured airflow and so on until the branch with the least percentage of measured airflow is regulated.
20. All branched airflow to be regulated so that each terminal on the entire system will be handling the same percentage of design airflow as every other terminal in the entire system. Finally the main airflow is adjusted to the required design flow.
21. Verify fan capacity and operating conditions again
22. Make a final adjustment in the fan drives if necessary.
23. Prepare all reports and submit as required, example is shown in Figure A7.


YOUNG'S ENGINEERING CO LIMITED
TESTING & COMMISSIONING RECORD - VAV TERMINAL UNIT

Project Name: PS 2 Rm   Ref.No.: AC-01   Page No.: 1 of 8
Scope of Work: MVAC Installation   T&C Date: 10/8/09
Location of Works: PS 2 Rm 5/F
Contract Specification: NA
Shop Drawings: VAL 90651 /AC/01   Ventilation mode: Mode 1 (300l/s, 400l/s)

1. Unit Data
Manufacture: Johnson Control   Country of Origin: PRC
Elect. Characteristic: Voltage: 220V   Phase: 1   Hz: 50Hz

2. Performance Test

VAV Nos. Please refer to layout	Function Check	Elect. Heater Running Amp		Maximum Air Flow Rate (L/s)		Minimum Air Flow Rate (L/s)		Comments
		Step1	Step 2	Design	Actual	Design	Actual	
VAV-1	ok.	NA	NA	75	82	50	42	
		NA	NA	75	77	50	46	
		NA	NA	75	74	50	45	
VAV-2	ok.	NA	NA	75	76	50	47	
		NA	NA	75	77	50	47	
		NA	NA	75	71	50	41	
				450	457 (≥ 402)	200	268 (≥ 184)	

Note: 1. VAVs of the same model and design characteristic are only sampling tested.
2. Minimum Air Flow Rate= NA % of Maximum Air Flow Rate
3. After commissioning, thermostat is set back to design temperature 24 °C

3. Comments:
Passed   Failed

Reported by: Date: 10/8/09
Inspected by: Date: 10/8/09
Witnessed by: Date:

Doc.NO. WI-TCM-AS-14   Issue No. 1 Rev.No. 0   16 November 1995

Figure A7: Testing & commissioning report (sample)

A2.3 Testing of different ventilation modes

1. Select ventilation modes of this air distribution system.
2. Check Damper positions in accordance with damper Modes table of respective mode condition.
3. Complete Testing and commissioning all record sheet
4. Select different modes system as shown in Table A1 and repeat above items.

Table A1 Mode Table of various air distribution methods

Ventilation Mode	Motorized on-off dampers operation (Location of each damper shall refer to Figure 2.3)										
	MD-01-S	**MD-02-S**	**MD-03-S**	**MD-04-S**	**MD-05-S**	**MD-06-S**	**MD-07-S**	**MD-08-S**	**MD-01-R**	**MD-02-R**	**MD-03-R**
1. mixing	Open	Open	Close	Close	Close	Close	Close	Close	Open	Close	Close
2. Displacement	Close	Close	Open	Close	Open	Open	Close	Open	Open	Close	Close
3. Stratum	Close	Close	Open	Open	Close	Open	Open	Close	Open	Close	Close
4. Modified-Stratum-1	Close	Close	Open	Open	Close	Close	Close	Open	Close	Open	Close
5. Modified-Stratum-2	Close	Close	Close	Close	Open	Open	Open	Close	Close	Close	Open
6. Modified-Stratum-3 (#)	Close	Close	Close	Open	Close	Open	Open	Close	Close	Close	Open

(#) Modified-Stratum-3 has been installed in Phase 2, Jan.201 by different funding and tested by same testing and commissioning processes with the pervious systems

A3 Quality Control during construction stage

All units and accessories had been fully conformed to the highest commercial standard and had been designed, constructed, rated and tested in accordance with an approved laboratory or authority. Materials were comply with the various British Standards, listed elsewhere in the specification of contractual document or other approved international standards. Factory applied acoustical and thermal insulation, including facing and adhesive, was to be fire-resistant and to conform to requirements of NFPA and HKSAR Fire Services Department. For example, Air handling units and accessories had been supplied by manufacturers experienced in the design and construction of similar equipment for at least five years as clearly stated in the contractual document. Before actual installation work, contractor furnished all of specified information for each unit/device/equipment to me for approval, such as wiring and control diagrams; manufacturer's shop drawings; complete manufacturer's printed catalogues; physical dimensions and operating weights; ductwork and pipework connection details; and mounting and fixing details, etc.

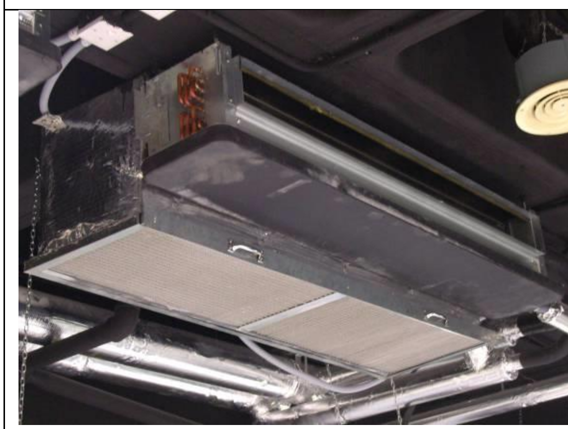
The following equipment has been successfully completed the predicted testing and commissioning process for conducting my research inside this environmental chamber.

1. Air side equipment (see Photo A37 to A40)
2. Water side equipment (see Photo A41)
3. Testing point (see Photo A42)
4. Building Management System (see Photo A43 to A44)
5. Electrical Services (see Photo A45 to A46)

Photo A37: Air side equipment 1

Air handling unit

Fan coil unit



Fan Coil Unit at ceiling

VAV box

VAV box (inside look)

VAV box at ceiling

Photo A38: Air side equipment 2

Exhaust air fan

Flexible duct

Rectangular duct

Damper control

600 mm x 600 mm square supply air diffuser

600 mm x 600 mm square return air grille

Photo A39: Air side equipment 3

	
Air plenum	Air plenum
	
Damper for air plenum	Damper control for air plenum with insulation
	
265 mm x 265 mm square type supply air diffuser	265 mm x 265mm square type dummy diffuser

Photo A40: Air side equipment 4

	
Pressure sensor	VAV terminator
	
AHU control panel	Frequency inverter

Photo A41: Water side equipment

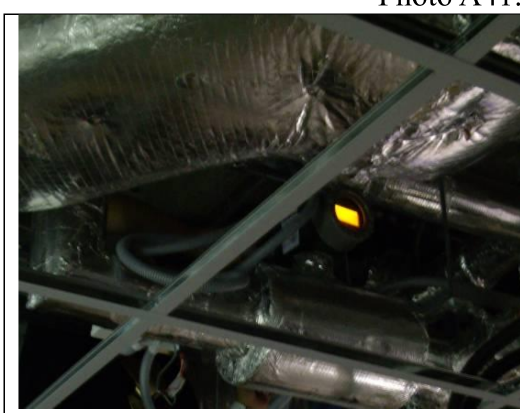
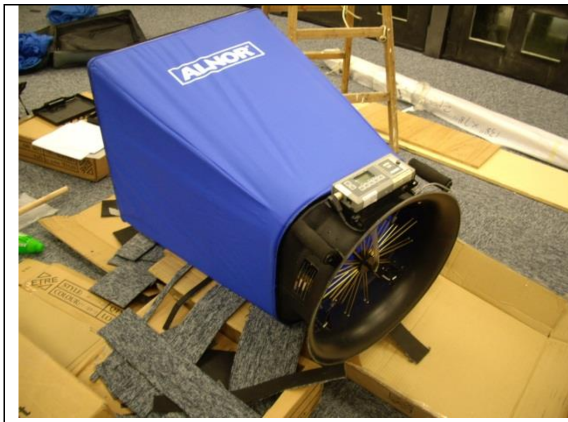
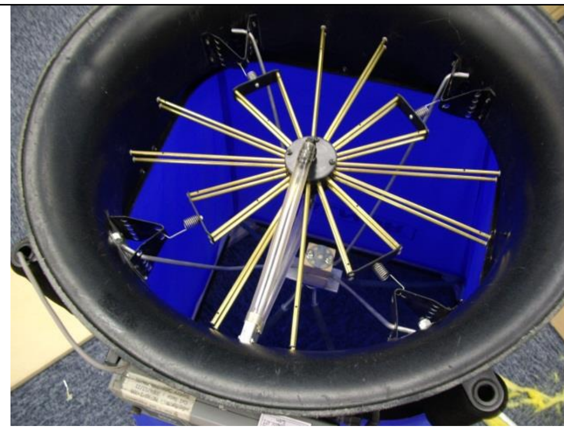
	
Flow meter	Chilled water pipe

Photo A42: Testing point by testing and commissioning equipment



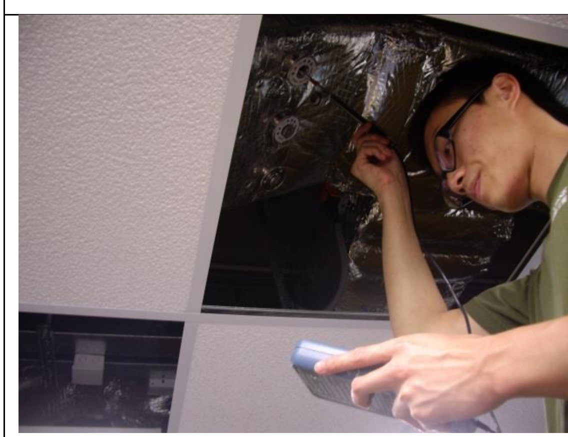
Equipment for air flow rate testing



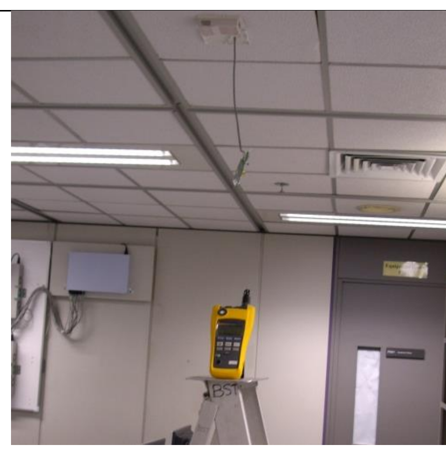
Top view of flow hood

Testing of air flow rate

Testing points



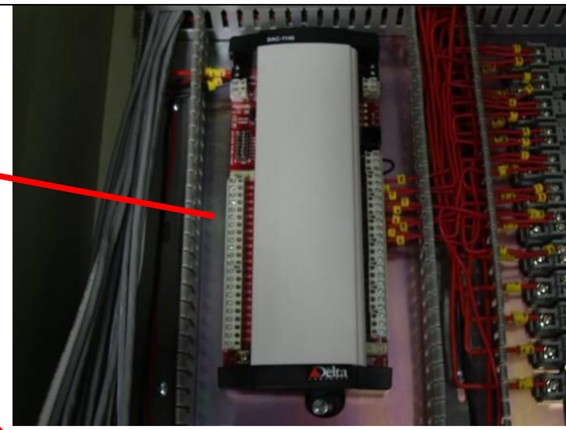
Testing of Air velocity



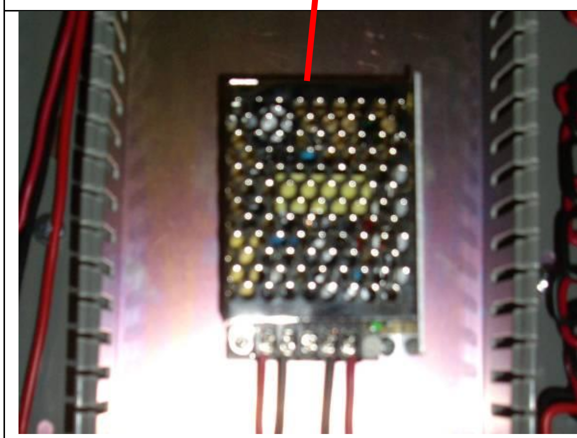
Testing the CO_2 and temp. sensor

Photo A43: Building management system – control equipment 1

DDC control panel

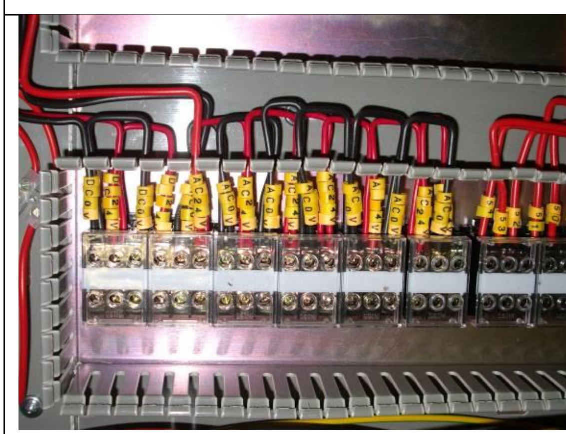


Digital direct control(DDC)



DC transformer

AC transformer



Wiring connection

Actuator of damper

Photo A44: Building management system – control equipment 2

	
Energy meter	Sensor mounted at ceiling
	
Temperature/RH sensor	Carbon dioxide sensor

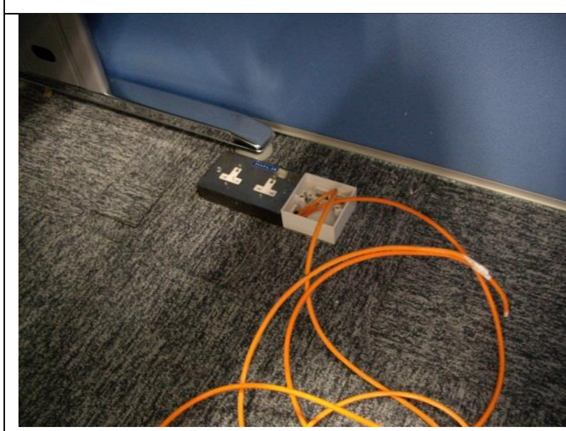
Photo A45: Electrical services 1

T5 tube lighting

Fitting

Dimmable DALI electric ballast

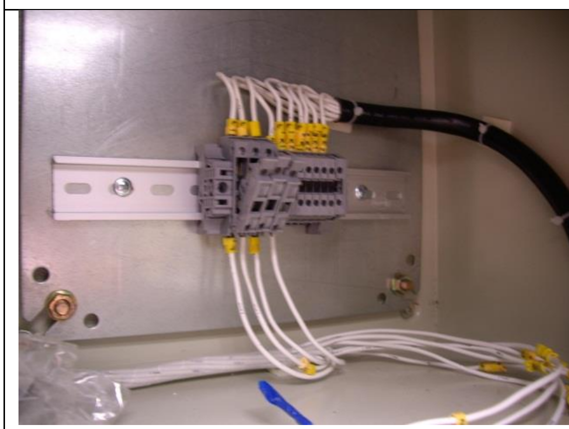
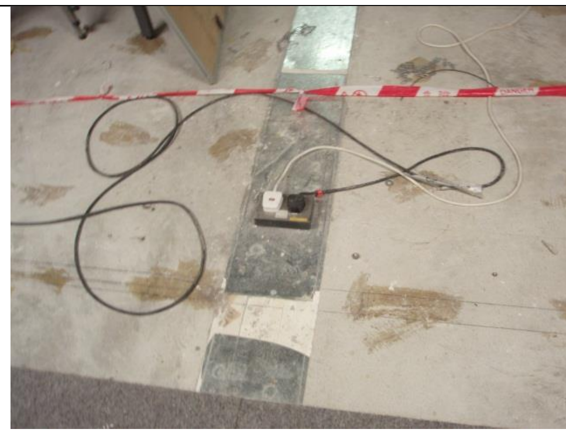
Category 6 cable for data transmission



Category 6 cable at floor

Category 6 cable at wall

Photo A46: Electrical services 2

	
Power meter	Power meter
	
Power meter	Cable run at raised floor
	
Socket outlet mounted at ceiling	Wiring inside control panel

APPENDIX B: AWARDS AND PUBLICATIONS

The environmental chamber is become a teaching, learning and research tool for undergraduate, master and PhD study to investigate any innovative and discovering air distribution strategy topics. The awards and publications are summarized in this Appendix.

B1 Awards

The list of successfully completed five (5) projects and won a competition using this environmental chamber in my local City University of Hong Kong are listed as follows:

B1.1 Funded projects (in the period of Month/Year)

1. 2/2012-1/2013, Development of a discovery-enriched teaching and learning system for engineering students of energy audit by in-class and out-class learning activities, Project number: 6000367, Teaching Development Grant (Principle investigator)

2. 9/2011-9/2012, Environmental sustainability "Headcount Index for CityU air conditioned area for engineering students in project studio 2, Project number: 6986002, Campus Sustainability Fund, Communities of Practice (Principle investigator)

3. 2/2009-4/2010, Full scale experimental investigation on comparison of the mixing ventilation system and localized ventilation system, BST Divisional Research Grant (DRG) for small scale projects (Principle Investigator)

4. 12/2007-6/2008, Course Teaching Material Development to upgrade the course material and develop new teaching and learning activities and assessment task for implementation in the OBTL approach. (Principle Investigator)

5. 3/2013-3/2014, Real-time mobile App to discover Energy Management Opportunity, DEC Mobile App Development Grant (Principle investigator)

B1.2 Student Award

6. 5/2014, As supervisor of Graduate- Mr. Lai Chun Tak (2011 cohort) won the first prize (Tertiary Institution Category) of Samsung Smart Education competition named "Samsung Solve for Tomorrow" contest by designing new energy-saving ventilation and air conditioning system and awarded a total HK$350,000 of cash and Samsung's product. The aim of this competition aims is to encourage students to use technology to solve environmental issues and create a greener HKSAR. His "Humanized Variable-Air-Volume Air Conditioning System" would automatically control the supply of fresh air based on collected data and could reduce an office's energy use by more than 10 percent during working hours.

B2 Publications

B2.1 Journal Articles

1. Cheng Y., Fong M.L., Yao, T., Lin Z., Fong, K.F. (2014). Uniformity of stratum ventilated thermal environment and thermal sensation. Indoor Air, Volume 24, Issue 5, October 2014, Pages: 521–532. Article first published online: 17 MAR 2014, DOI: 10.1111/ina.12097.

2. Fong, M.L., Lin, Z., Fong, K.F., Chow, T.T., Yao, T. (2011). Evaluation of thermal comfort conditions in a classroom with three ventilation methods, Indoor Air, 21(3), pp 231-239, 2011(Scopus).

B2.2 Conference Articles

3. Lin, Z., Fong, M. L., Wang, J. (2013). Stratum Ventilation — a Solution to Elevated Room Temperatures. The 11th REHVA World Congress and the 8th Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings, Proceedings of Clima 2013, 16/6/2013 – 19/7/2013.

4. Lin Z., Fong M. L. (2013). Design of Thermal Comfort in a Stratum Ventilated

Classroom. The 11th REHVA World Congress and the 8th Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings, 16/6/2013 - 19/7/2013.

5. Fong M. L., Leung M.C., Mok Y.K., TSE C. F., LAI L.L. (2012). New MVAC control by making use of Human Behavioral Based Technique to achieve Energy Efficiency. The 9th IET International Conference on Advances in Power System Control, Operation and Management (APSCOM 2012), 18/11/2012 - 21/11/2012.

6. Fong M. L., Chui H.L. (2012). Development of a Discovery-Enriched Teaching and Learning System for Engineering Students of Energy Audit. The 10th International Conference on Modern Industrial Training, pp 318-320, 20/10/2012 - 23/10/2012

7. Fong M. L., Tse C.M., Wong H.C., Man T.M. (2012). Intelligent Building System in Project Studio with energy saving case study by Headcount Index. The 12th International Conference on Clean Energy, 26/10/2012 - 30/10/2012.

8. Fong, M.L., Fong, K.F., Lin, Z. (2010). Experimental study of determining neutral temperatures for conventional mixing and stratum ventilation modes in environmental chamber, 5th Asian Conference on Refrigeration and Air-conditioning (ACRA 2010), 7/6/2010 - 9/6/2010.

9. Chow, T.T., Fong, K.F., Tse, N., Fong, A., Lin, Z., CHAN, A. (2010). Studying energy audit via intelligent project studio, Clima 2010, 9/5/2010 - 12/5/2010.

10. Fong, K.F., Fong, M.L., Chow, T.T. (2009). Measurement and verification for performance contract of HVAC energy conservation in HKSAR, Proceedings of the 4th Asian Conference on Refrigeration and Air-conditioning, 20/5/2009 - 22/5/2009.

- End -